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(54) Axial flow pump/ marine propeller

(57) An axial flow pump/marine propeller (10; 100; 200; 300; 400) has a pump body carrying a stator (20; 120; 220; 320; 420) which defines a cylindrical inner space, a rotor member (40; 140; 260; 360; 460) being rotatably located in the inner space, and electromagnetic means (24, 26, 44; 124, 126, 144) arranged between the stator and the rotor for generating a rotational magnetic field to drive the rotor to rotate. The rotor has a hollow structure with one or more propelling blades (48; 148; 283; 366) for driving the fluid to flow along the central axis of the pump. This arrangement provides a driving force which produces minimum fluid disturbance and drag therefore a high energy efficiency. The rotor is supported in the pump body by a suspension bearing mechanism which provide rotational and thrust bearing in response to the rotor's rotational and/or axial movements to retain the rotor at a balanced position. In the disclosed embodiments, the bearing means can be formed by mechanical (50; 380; 480), magnetic (150; 190; 240; 250) or hydraulic (49; 149; 270; 356) mechanisms, or a combination of any of them, which keep the rotor fully suspended in and coaxially aligned with the pump body during its operation while at the same time reduce the friction and wear of the rotating parts. By using the magnetic and/or hydraulic suspension to support the rotor, it also avoids the problems and costs of having sealing and lubricating arrangements for the rotary parts, therefore reduce both the manufacturing and maintenance costs. Further arrangements (160, 166) are incorporated into the structure to automatically adjust the axial position of the rotor when it is fully suspended by the hydraulic and/or magnetic mechanisms. Compressible spiral blades can be used to cope with frequent changes of electric inputs and/or output loads. Some of the disclosed embodiments provide clog-free structures suitable for a wide range of applications, such as sewerage pump, sludge pump or marine propeller. An arrangement of two anti-spiral driving members (366, 370; 466, 470) are disclosed for high pressure applications.

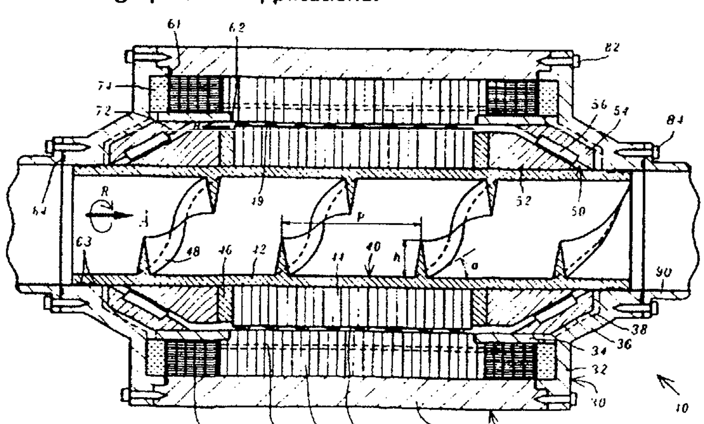

Fig 1

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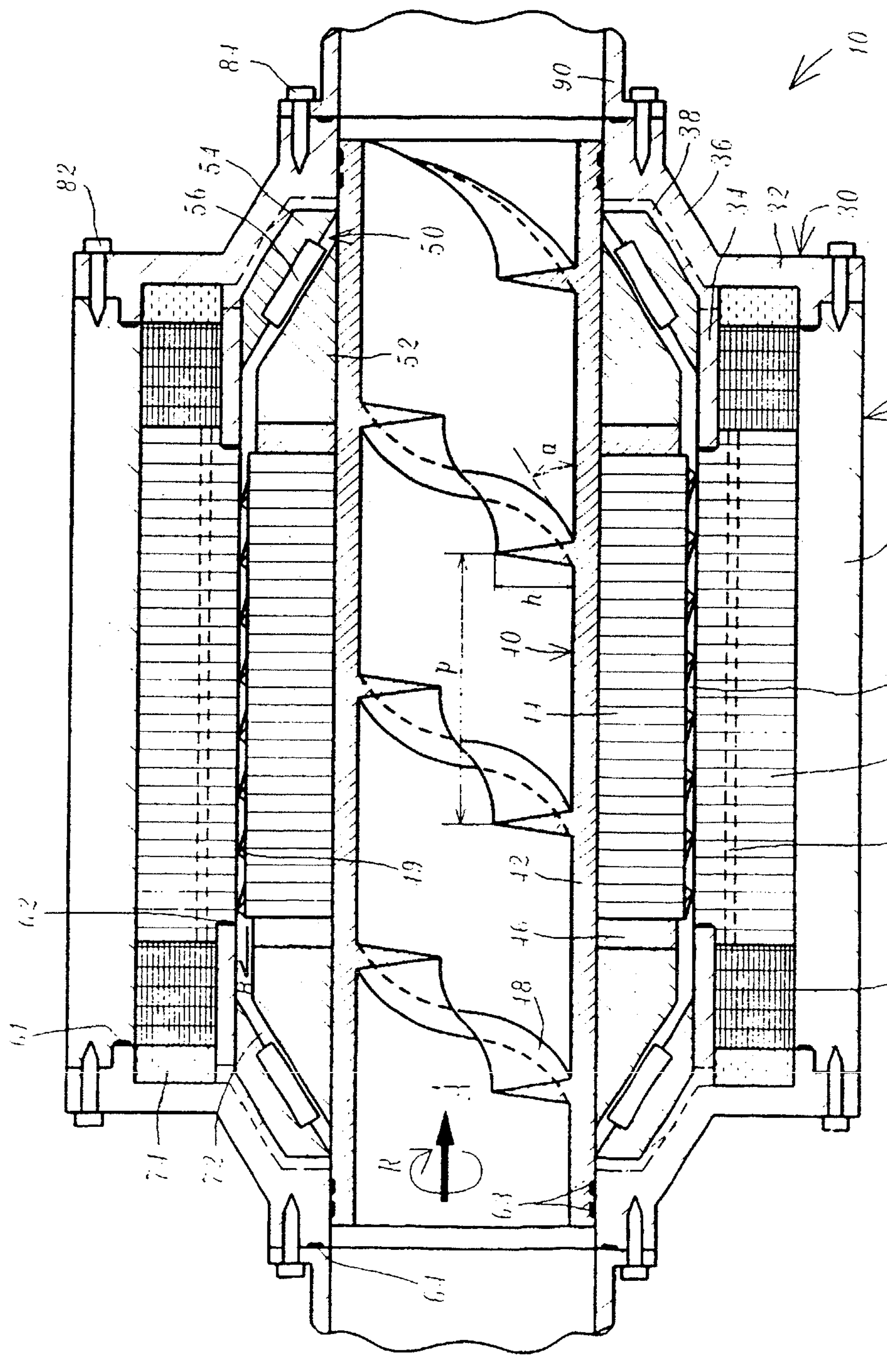
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Fig 1

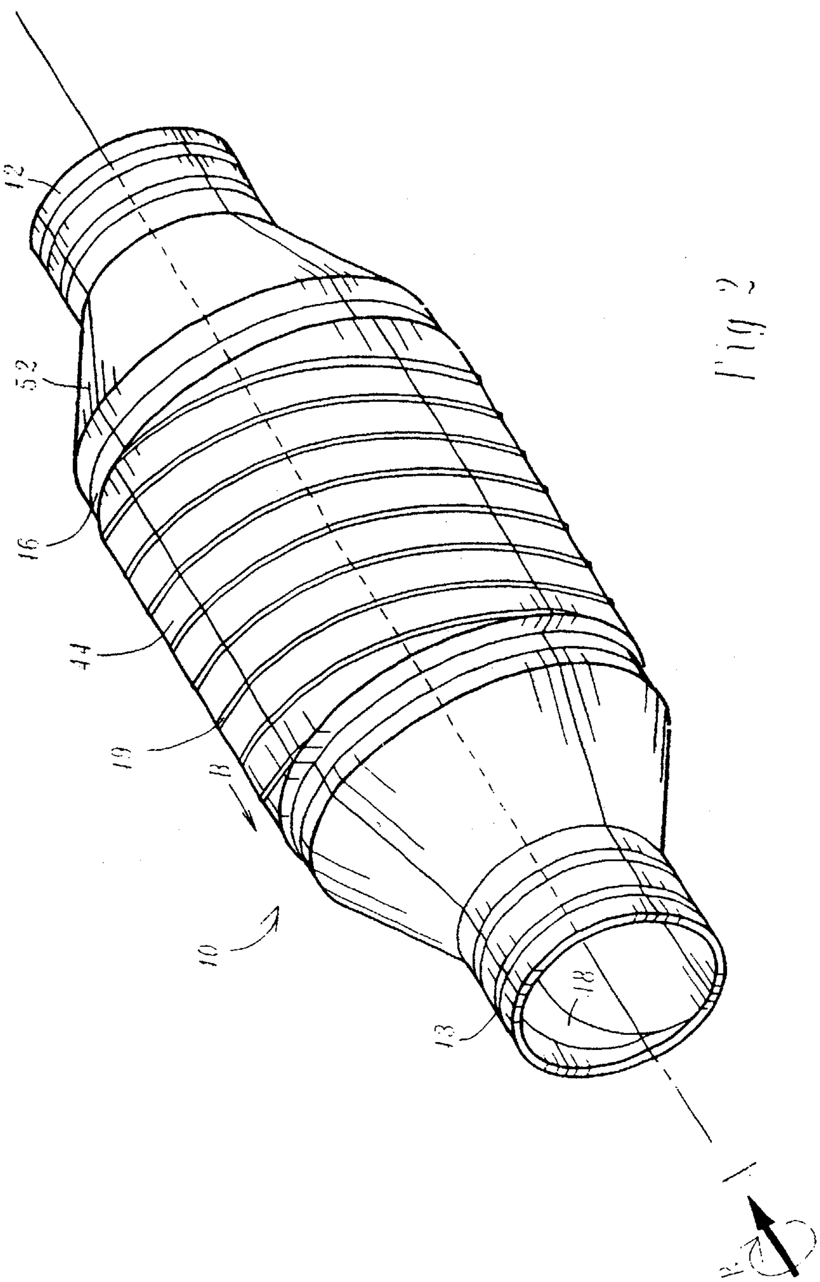

Fig 2

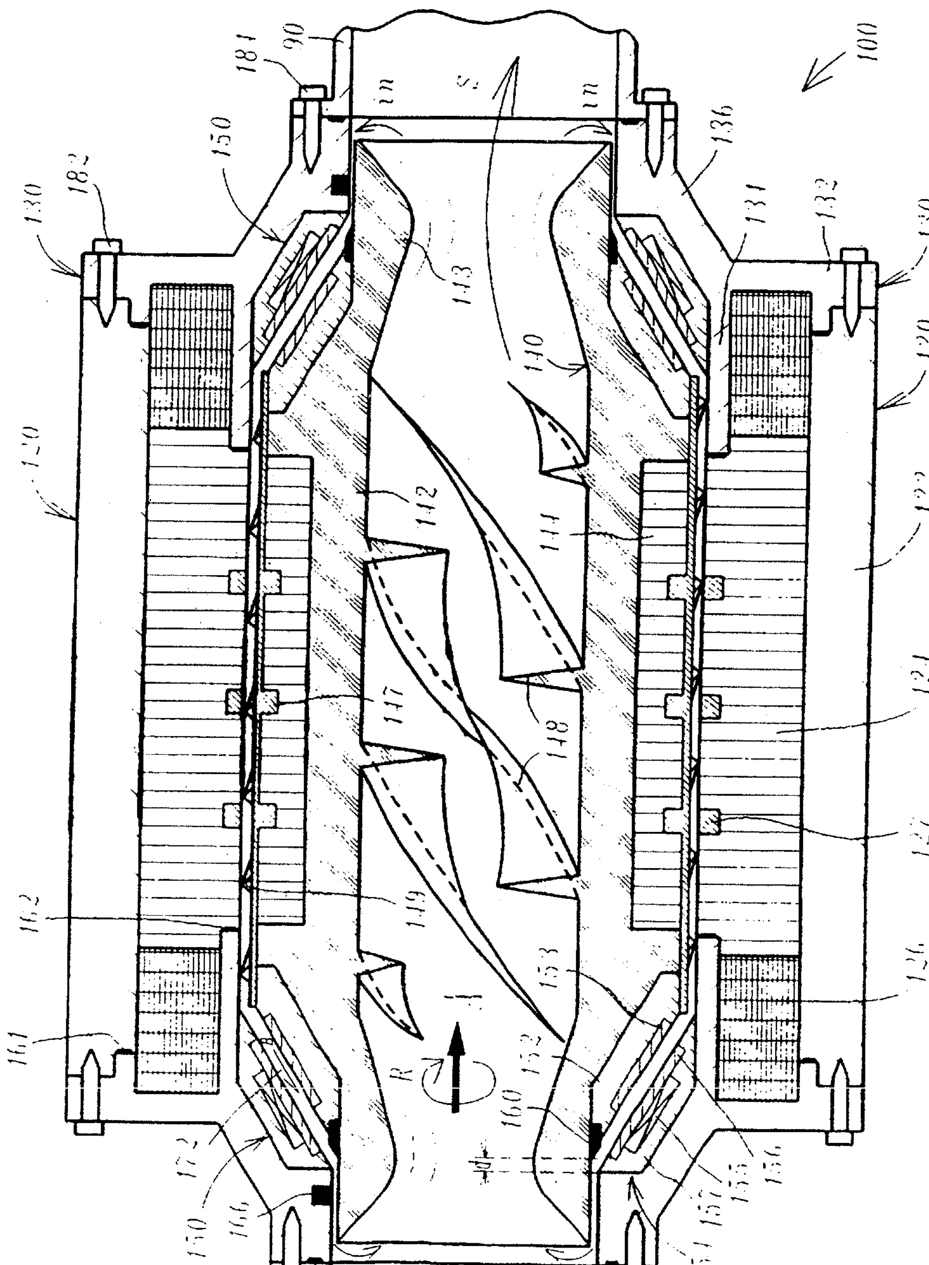

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Fig 3

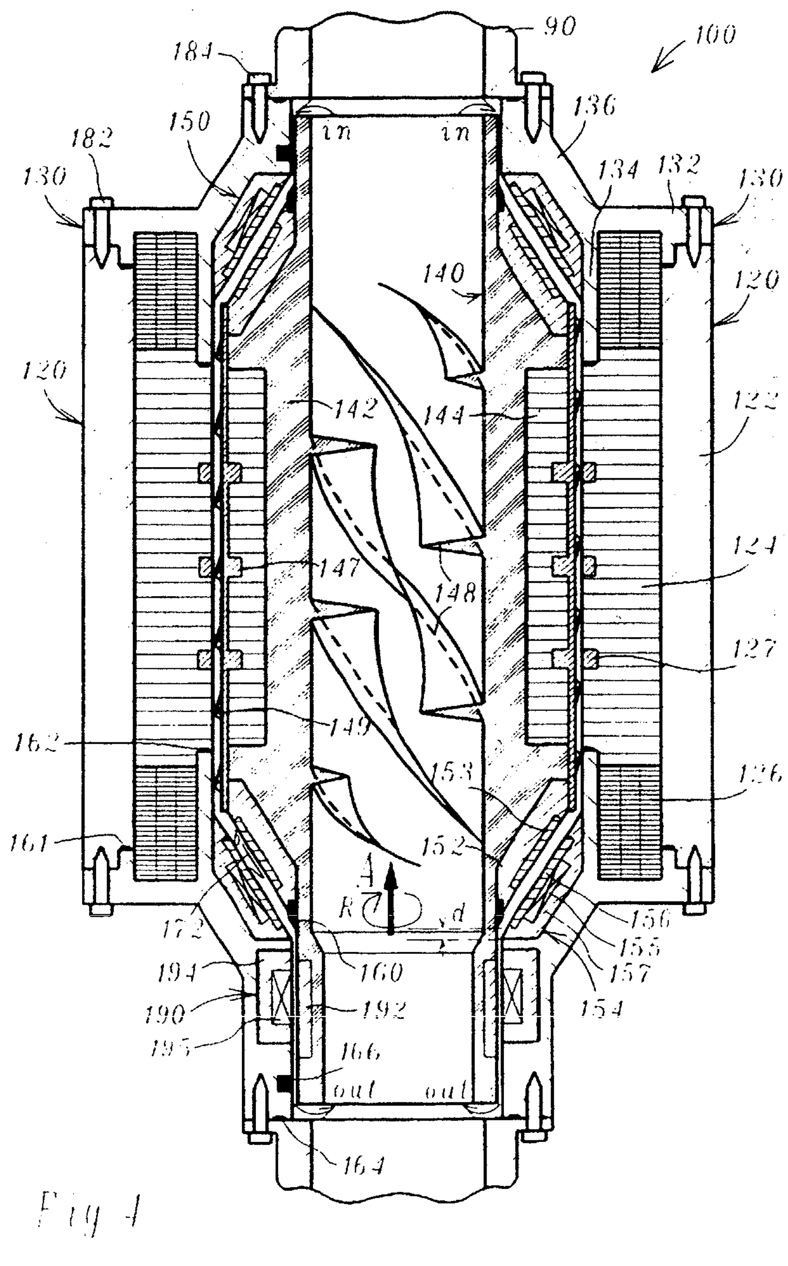
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Fig 4

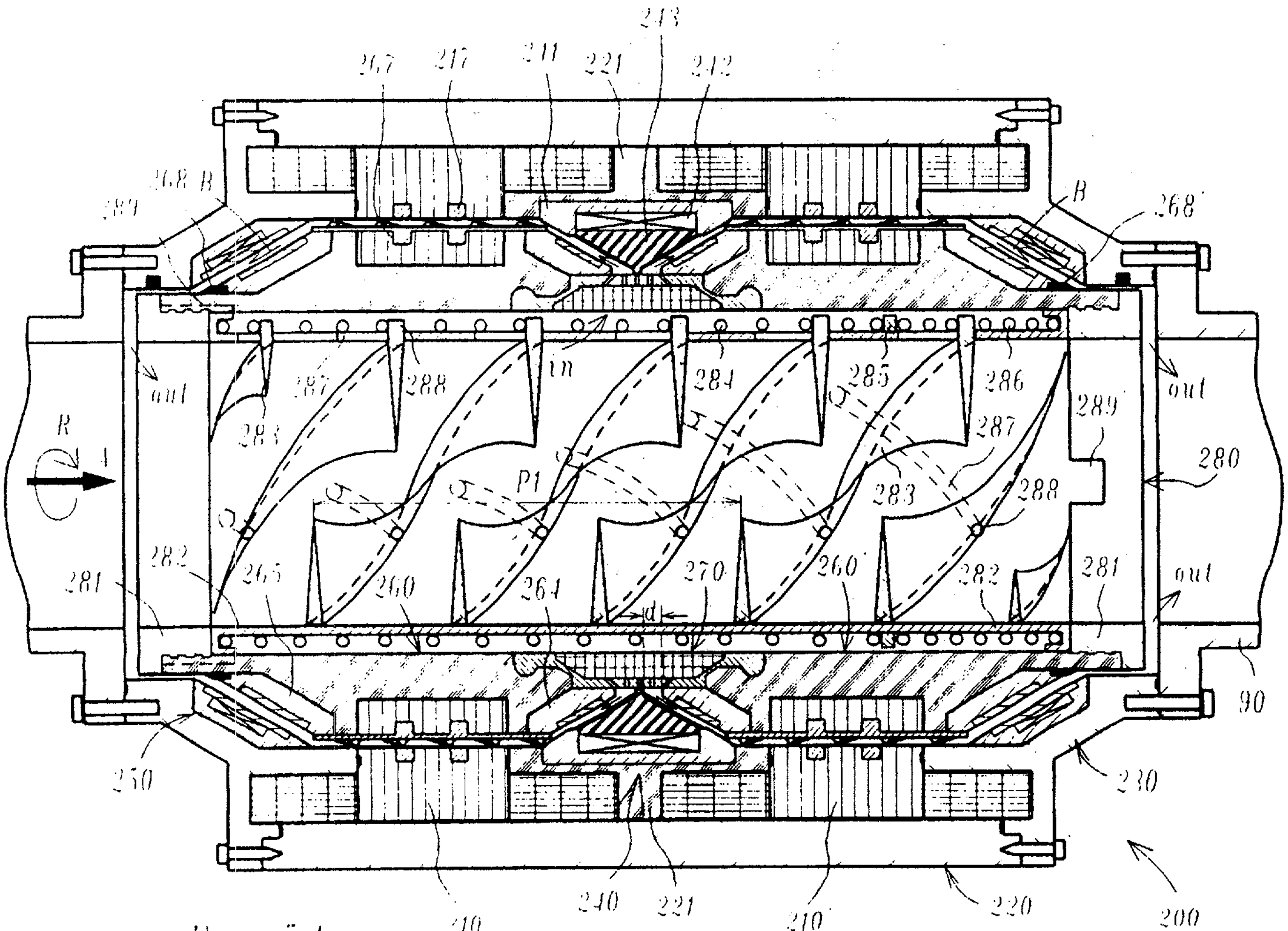
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Fig 5A

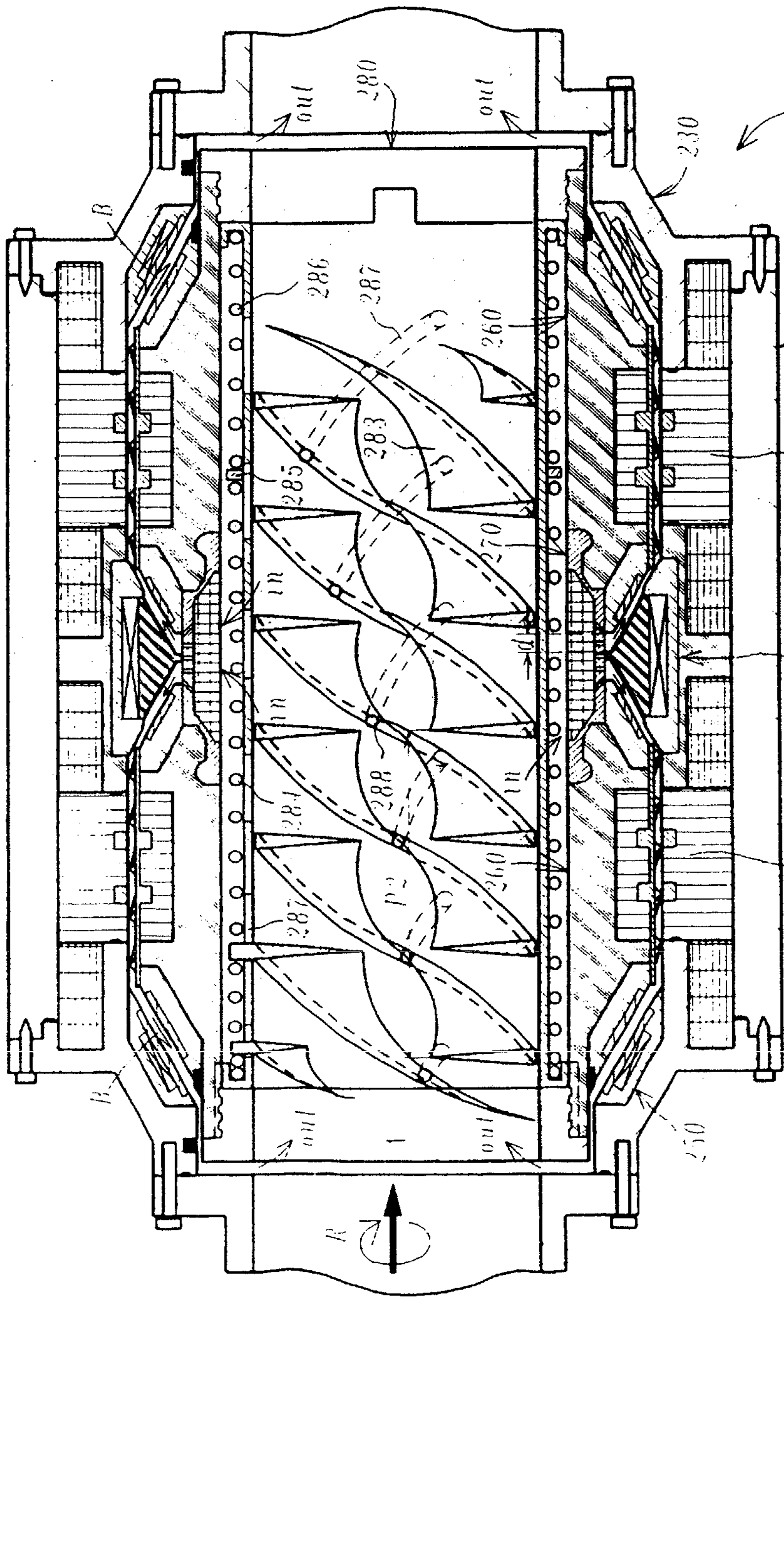
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Fig 5B

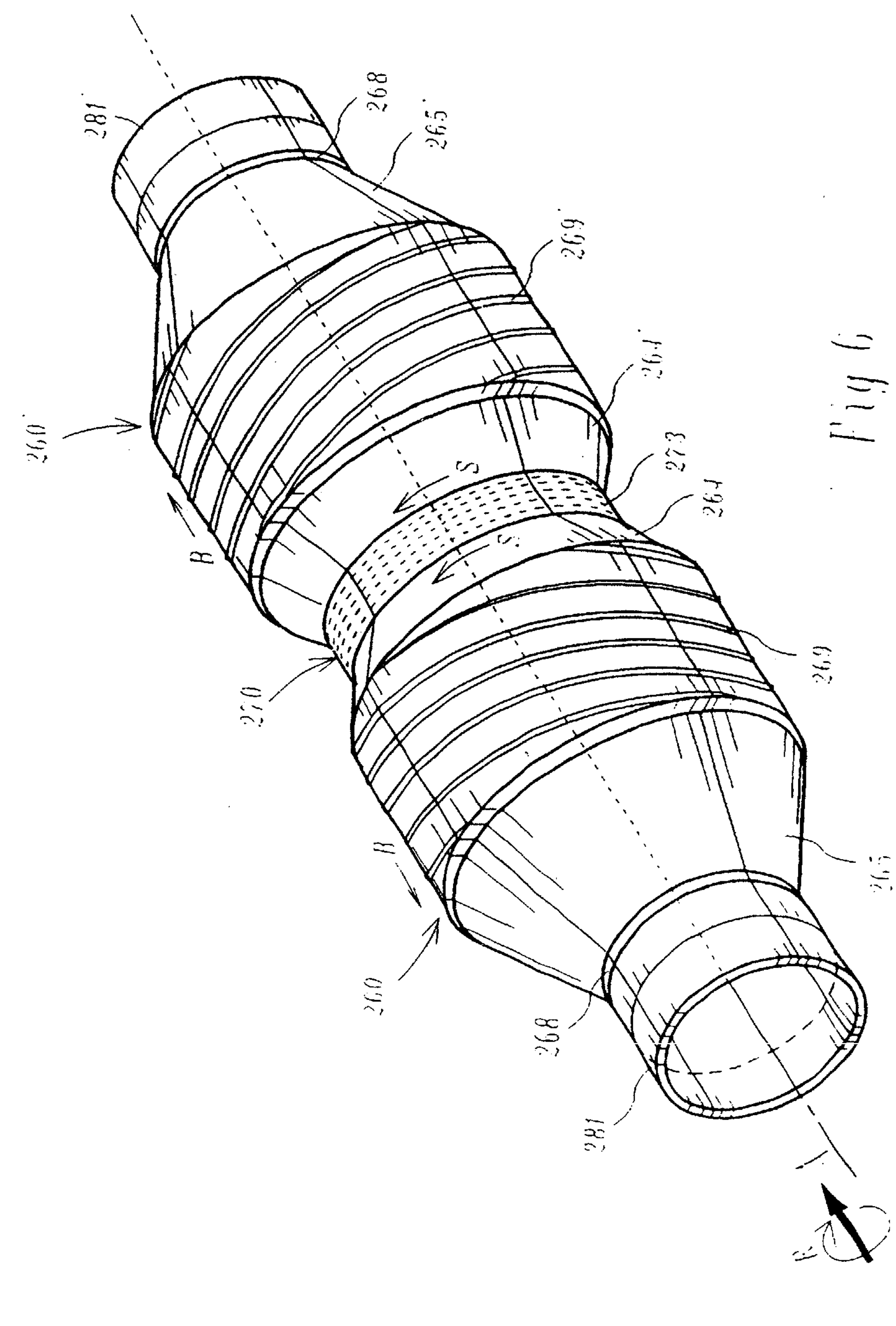

Fig 6

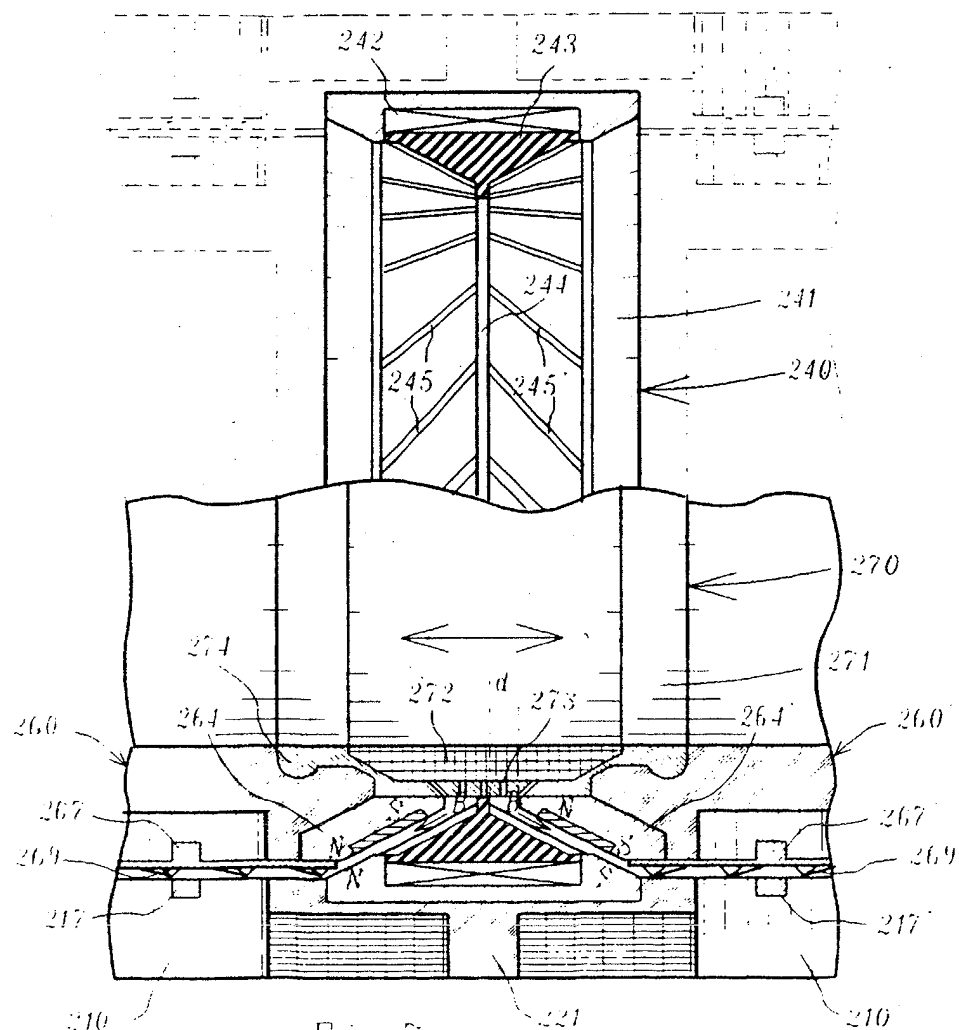
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Fig 7

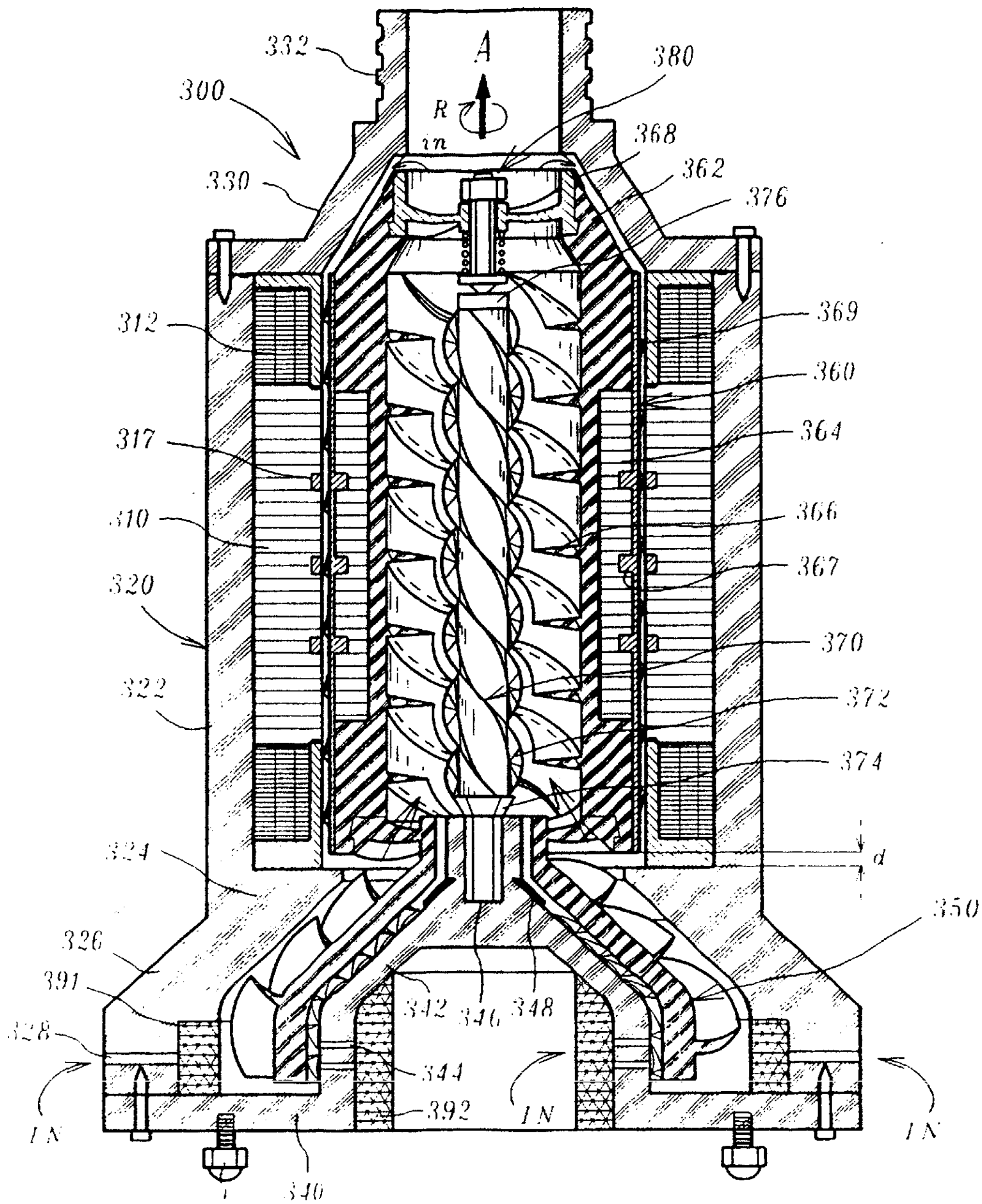
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Fig. 8

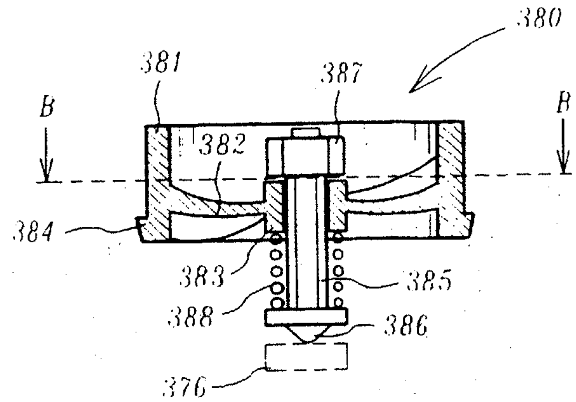

Fig. 8A

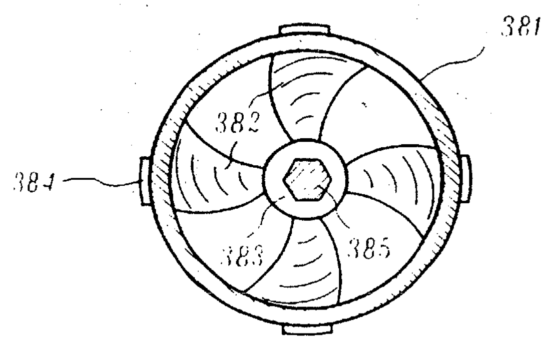

Fig. 8B

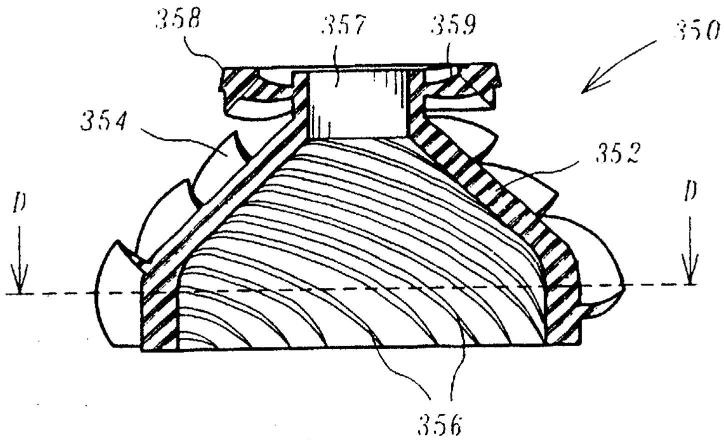

Fig. 8C

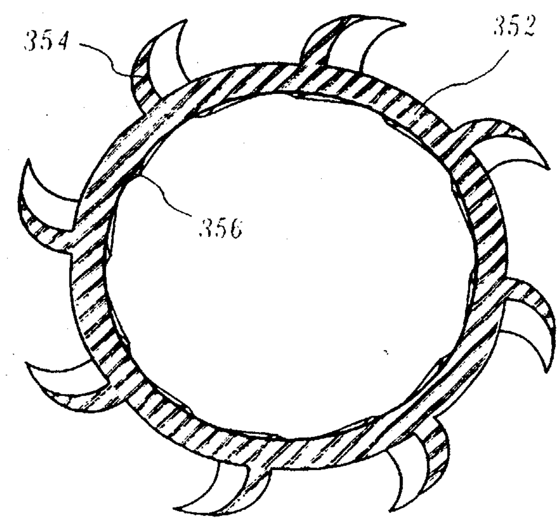

Fig. 8D

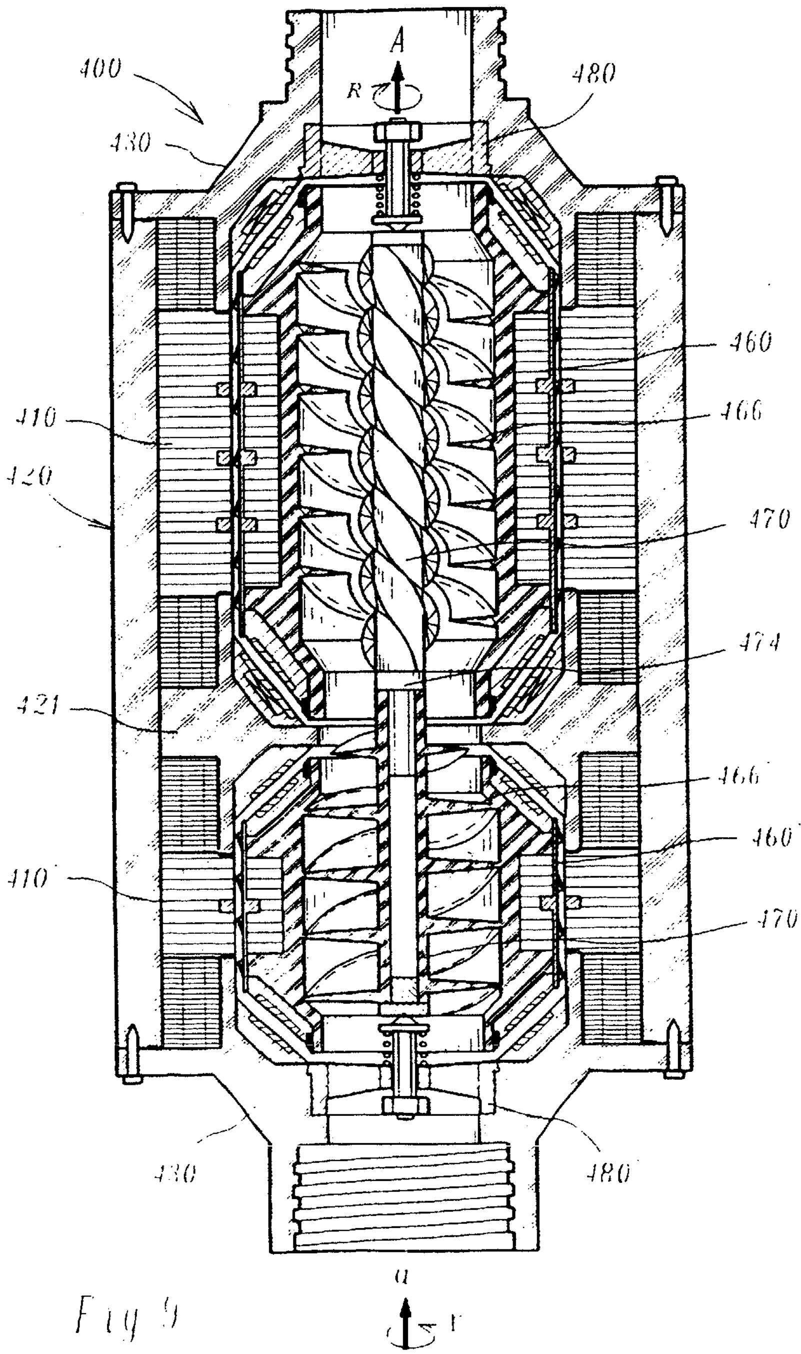
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Fig 9

2301399

Axial Flow Pump/Marine Propeller

Technical Field of Invention

The present invention relates to an axial flow pump and/or marine propeller for forming a fluid axial flow, and more particularly, to an axial flow pump and/or marine propeller having a built-in electric motor as an integrated part of the pump or propeller.

Background of Invention

It is known in the art that an axial flow pump having a propeller driven by a motor can produce an axial flow in a pipeline. This is an efficient way of forming a steady fluid flow. However, in a conventional arrangement the pipeline has to be angled so that the propeller can be fitted into the pipeline with the motor shaft extending through the wall of the pipeline at the angled position, otherwise a more complicated gear arrangement instead of a direct shaft connection has to be used to transfer the rotational driving force of the motor to the propeller. Such arrangements make it difficult to arrange a pipeline to suit a pump, while on the other hand, a reliable sealing of the rotary shaft which passes through the wall of the pipeline is difficult to achieve but easy to wear out. When such a pump is used for marine propelling, it is very easy to get the propeller tangled with, for example, seaweed or other foreign objects which enter the inlet of the pump by its sucking force.

Summary of Invention

It is, therefore, a main object of the present invention to provide a pump or propeller which overcomes the above problems and disadvantages.

According to one aspect of the present invention, there is provided an axial flow pump/marine propeller comprising: a pump body with two open ends, a stator member carried by the body and defining a cylindrical inner space extending between its two open ends, a hollow rotor member with means for propelling fluid therethrough being rotatably fitted in the inner space, and electromagnetic means arranged between the stator and the rotor for generating a rotational magnetic field to drive the rotor to rotate; wherein the rotor member is supported in the pump body by a suspension bearing arrangement which provides rotational and thrust bearing when the rotor rotates. It is advantageous that the bearing

arrangement includes means sensitive to the rotor's rotational and/or axial movements to retain it at a balanced position during its operation.

Preferably, the suspension bearing arrangement includes at least one spiral vane formed on the outer surface of the rotor for producing a peripheral flow of a fluid in the gap between the stator and the rotor when the rotor rotates.

According to another aspect of the present invention, there is provided a pump with bearing means formed by mechanical bearing arrangements fitted to both ends of the rotor which also has sealing arrangements for keeping a lubricating/cooling medium in a gap between the rotor and the stator, with which gap sealed from the axial flow in the fluid passage.

According to yet another aspect of the present invention, there is provided a suspension bearing mechanism for keeping a rotary member levitated during its rotation.

Advantageously, the suspension bearing mechanism includes a magnetic axial registration mechanism and/or a magnetic axial suspension mechanism formed by a combination of an annular electromagnet and a matching annular permanent magnet.

Also advantageously, the suspension bearing mechanism includes a fluid suspension system with flow dividing means for keeping the rotary member self-balanced during its rotational operation.

According to yet another aspect of the present invention, there is provided an axial flow pump/marine propeller having fluid driving means formed by one or more flexible spiral blades which can be compressed in axial direction in response to the changes of power input or working load.

According to yet another aspect of the present invention, there is provided an axial flow pump/marine propeller having two co-axially arranged fluid driving members with spirally arranged fluid driving means in opposite spiral directions so that when they have relatively opposite rotational movements around the common axis, they cancel each other's swirling effects to produce a high pressure axial output. Advantageously the central driving member is supported by a pivot bearing arrangement. Also advantageously a conical impeller member is fitted to the upstream end of the pump to further promote the axial flow.

According to yet another aspect of the present invention, there is provided an axial flow pump/marine propeller which can be used as an electric generator and or flow-meter.

For the sack of easy understanding, the axial flow pump/marine propeller of the present invention is to be described hereinbelow as a pump unless specifically explained otherwise.

Brief Description of Drawings

The above described objects, aspects, features, advantages, and further structural and functional details of the present invention will become more readily apparent from the following description of the preferred embodiments with reference to the accompanying drawings, in which:

Fig. 1 is a sectional view illustrating a first preferred embodiment of the pump/marine propeller according to the present invention;

Fig. 2 is a perspective view of the rotor 40 used in the first embodiment of the present invention;

Fig. 3 is a sectional view of a second preferred embodiment of the pump/marine propeller according to the present invention;

Fig. 4 is a sectional view of a third preferred embodiment of the pump/marine propeller according to the present invention;

Figs. 5A and 5B are sectional views showing a fourth preferred embodiment of the pump/marine propeller according to the present invention;

Fig. 6 is a perspective view of the rotor assembly used in the fourth embodiment of the present invention;

Fig. 7 is an enlarged sectional view showing details of a localised portion of the fourth embodiment of Figs. 5A and 5B;

Figs. 8 to 8D show a fifth preferred embodiment of the present invention; and

Fig. 9 is a sectional view showing a sixth preferred embodiment of the present invention.

Detailed Description of Preferred Embodiments

As shown in Figs. 1 and 2, a pump 10 according to a first embodiment of the present invention has, generally speaking, a pump body formed by a cylindrical stator 20 and two generally annular end cap members 30, each fitted to one end of the stator 20, for defining together with the stator a cylindrical inner space in the pump body. A rotor 40 of a generally

tubular structure is fitted in the inner space. Each cap member 30 has a central opening which is coaxial with the cylindrical stator 20 and the cylindrical rotor 40 so that an axially extending fluid passage is formed along the rotational axis of the pump 10. Inside the pump body, the rotor 40 is supported at each axial end thereof by a bearing arrangement 50, which keeps it coaxial with the pump body and provides rotational and thrust bearing when the rotor rotates. It is also shown in Fig. 1 that the pump 10 is connected to two tubes 90 to form an integral part of a pipeline.

More specifically, the stator 20 has an outer housing 22, a magnetic core 24 with electric windings 26 and through bores 28 indicated by dash lines, the function thereof is to be explained below. A magnetic gap 72 is formed between the cylindrical inner surface of the magnetic core 24 and the outer surface of the rotor 40. The arrangement of these members is made to generate, when electric currents are supplied to the windings 26, a rotational magnetic field, as in a conventional electric motor. Conventional arrangements can be used here for this purpose and there is no need for further detailed description.

The two cap members 30 are of the same general structure, i.e. having an annular flat wall part 32, an annular slope wall part 36 which defines a central opening, and a sleeve part 34. Each cap member 30 is secured at its outer edge to the stator 20 by a number of fast members 82, while at the inner edge near the central opening it is secured to a tube 90 by fast members 84. On the inner surface of the slope wall part 36, there are a number of channels 38 extending radially from the inner edge of the central opening of the cap 30 to and passing through the sleeve member 34, to enter a chamber 74 formed between the cap and the stator, their function is to be explained below. The two chambers 74 are in fluid communication with each other via the through bores 28 in the magnetic core of the stator 20, and with the gap 72 via the channels 38. Sealing members 61, 62 and 64 are used to ensure that the engagement positions between the cap member 30 and the stator 20 and that between the cap members 30 and the tubes 90 are fluid-tight, so that the connection between the pump and the pipeline is leakage proof and is not subject to any wear caused by a moving component.

As also shown in Fig. 2, the rotor 40 has an inner tubular member 42 which carries a magnetic core 44 on its outer surface secured between two supporting members 46. The magnetic core 44 can be constructed in the same way of that of a rotor of a conventional

electric motor, such as a squirrel cage motor, (the conductors of a squirrel cage are not shown in Fig. 1 or Fig. 2) so that it can rotate under the influence of the rotational magnetic field generated by the stator 20. The tubular member 42 has on its inner surface an propelling member 48 in the form of a screw blade, so that when the rotor 40 rotates the propelling blade 48 drives the whole volume of the fluid in the central passage to flow along its axial direction, as shown by the arrow A in Fig. 1. On the outer surface of the rotor 40, there are a number of spiral vanes 49 formed by cutting the surface of the magnetic core 44, or by adding a sleeve member with the vanes formed on its surface to cover the magnetic core 44, and this member can be made of an elastic material. The function of the vanes 49 are to be further explained hereinbelow.

The rotor 40 inside the pump body is supported by two mechanical bearing arrangements 50. Each bearing 50 has an inner bearing member 52 fitted to each end of the rotor 40, an outer bearing member 54 fitted to the cap 30 between the sleeve part 34 and the slope wall part 36, and a number of rollers 56. The purposes of the bearings 50 are to provide both thrust bearing and rotary bearing to the rotor 40 to keep it coaxial with the stator and to ensure a smooth rotation of the rotor. At each axial end of the rotor 40, further sealing members 63 are fitted in the seal grooves 43 (shown in Fig. 2) to provide a fluid-tight contact between the rotor 40 and the cap member 30, which forms the only movable sealing contacts in the pump. This arrangement ensures that the magnetic gap 72 between the stator 20 and the rotor 40 is sealed from the main-stream axial flow in the pipeline indicated by the arrow A, so that no dirt carried by the flow can enter the gap 72. In case a corrosive fluid is transferred in the pipeline, effective sealing arrangements also protect the other components of the pump from being corroded since only the cap members 30, the seals 63 and the inner surface of the tubular member 42 and the propelling blade 48 are in direct contact with the corrosive fluid.

A lubricating-cooling arrangement is formed between the rotor 40 and the inner surfaces of the pump body by filling the gap 72 with a lubricating-cooling medium. Since the gap 72 is in fluid communication with chambers 74, as explained above, the same medium can circulate via the channels 38 and the through bores 28 in the magnetic core 24 so as to work as lubricating fluid to the outer surface of the rotor 40 and the bearings 50 and as cooling fluid to the windings 26 and the magnetic core 24. As also shown in Fig 2, spiral vanes 49 are formed on the outer surface of the magnetic core 44 of the rotor 40 and their

spiral direction is opposite to that of the inner blade 48, these outer vanes 49 work in the same way as that of a vane pump and have the effect of driving the medium in the gap 72 to flow in a direction opposite to that of the arrow A, so as to flow through the channels 38 into the chamber 74 at the left end of the pump. This flow of the medium continues through the bores 28 formed in the stator's magnetic core 24 to the other chamber until it returns to the gap 72 from the other end of the pump. This circulation of the medium forms a peripheral fluid cushion around the rotor, which helps to keep the rotating rotor 40 well lubricated while at the same time keep the windings 26 and the magnetic core 24 cooled, because the heat picked up by the medium is easily transferred to the main-stream axial flow through the caps 30 and the tubular member 42 of the rotor 40. Conventional transformer oil can be used as the medium for this purpose. It should be noted that the peripheral cushion of the medium also serves as a hydraulic bearing to help keeping the rotor coaxial with the cylindrical inner surface of the stator magnetic core 24 and to provide a counter-thrust to the rotor, which reduces the burden of the mechanical bearings 50. To enhance the hydraulic bearing effects, spring biased one-way valves (not shown) can be fitted, e.g. in the chamber 74 to block the oil flow into it so that only when the pressure of the oil in the gap 72 is built up by the rotation of the rotor 40 to a value higher than this biasing force, i.e. when the rotor 40 rotates beyond a certain speed level, the circulation of the oil starts. A small oil supply tank (not shown) can be fitted to the pump body for supplying oil into this lubricating-cooling system to compensate any leakage into the main-stream flow via the seals 63, for this purposes, a stable oil pressure need to be maintained in the tank, e.g. by a spring loaded piston. Such arrangement is known in the art and do not need further description.

The operation of the pump 10 is to be explained further hereinbelow. When the stator windings 26 are connected to an electric power supply and the power is switched on, a rotational magnetic field is formed between the stator and the rotor which drives the rotor 40 to rotate. As shown in Fig. 1, when the rotational direction of the rotor is that shown by the arrow R, the flow direction is that of the arrow A. The blade 48, when rotating with the rotor 40, forces the fluid in the central passage to flow in its axial direction while at the same time the rotor itself is subject to a counter thrust force in an opposite direction. The fluid flow direction is reversible by controlling the electric power supply, hence the rotational direction of the magnetic field. On the other hand, as the rotor 40 rotates, the spiral vanes 49 on the outer surface of the rotor 40 would produce a peripheral flow of the lubricating-

cooling oil in the direction as shown by the arrow B, which is opposite to that of the axial flow inside the rotor because their spiral direction is opposite to that of the blade 48, therefore it always provides a counter thrust which helps to balance the rotor's radial and axial position and reduce the thrust force on the bearing. It should be noted that when the rotor's rotating speed is increased, i.e. when the rotor is subject to an increased thrust force, the vanes 49 would produce an increased peripheral flow with an increased balancing effect. That is to say the arrangement is sensitive to the changes of the rotor movements and it provides self-balancing effects which compensate such changes automatically. The back flow of the lubricating oil also keeps the oil pressure higher at the upstream side (left-hand side in Fig. 1) of the axial flow, with the effects of preventing any dirt from passing through the seals 63 and entering the gap 72 , and also keeping the seals 63 well lubricated so as to reduce their wear. In this arrangement, as long as the seals 63, especially those at the upstream side which are more subject to the thrust force of the axial flow, are intact, the pump would be able to operate normally because all the other components are less eligible to be worn out during the pump's operation.

Fig. 3 shows a pump 100 according to a second preferred embodiment of the present invention. The general structure and the operation principles of the pump 100 are similar to that of the first embodiment, therefore only the different features of the second embodiment are to be described hereinbelow.

As shown in Fig 3, the stator 120 and the rotor 140 are arranged to form a brushless DC motor, with an array of permanent magnets 144, preferably rare earth magnets, fitted on the rotor. This makes it suitable for applications requiring easy control of both rotational speed and direction, such as in case of a marine propeller.

The rotor 140 is fitted in the pump body as a "free" rotor, in the sense that during its operation, the rotor is fully suspended and levitated by magnetic and hydraulic forces without direct physical contact with any support member. At each end of the rotor 140, there is a space of a distance "d", as shown in Fig. 3, which allows the rotor to have a limited axial movement when it is fully suspended. This fully suspended or "floated" status of the rotor is maintained by the effects of one or more of the following suspension bearing mechanisms.

Firstly, a magnetic axial registration mechanism is formed by having a number of annular non-magnetic or high magnetic resistance members 127 on the inner surface of the

stator's magnetic core 124, and the same number of similar members 147 on the outer surface of the rotor, and the two sets of annular members 127 and 147 match each other when the stator and the rotor are in axial alignment. These annular members separate the magnetic coupling between the stator and the rotor into separated zones so that a maximum magnetic coupling can be achieved only when the zones have a perfect axial registration, which forms a neutral position, as shown in Fig. 3. Any axial movement of the rotor will cause misalignment of these magnetic zones therefore increase the total magnetic resistance in the magnetic circuit formed by the stator and the rotor hence reduce the magnetic flux in the circuit. In this case, the mechanism is highly sensitive to any axial movement of the rotor and an axial magnetic force would be generated between the stator and the rotor which tends to return the rotor to its neutral position. When the magnets 144 are of strong rare earth material, they produce strong registration force against any misalignment of the rotor away from its neutral position relative to the stator even the windings 126 of the stator 120 are switched off.

Secondly, each magnetic suspension bearing arrangement 150 has a conical ring shaped electromagnet member 154 fitted in the cap 130 and a matching conical ring shaped permanent magnet 152 fitted to an end of the rotor 140. The electromagnet member 154 has a magnetic core 157 with a generally U-shaped cross-section, a toroidal coil 155 fitted in the channel of the core 157 and a non-magnetic member 156 covering the coil in the core. When there is an electric current flowing in the coil 155, a magnetic flux forms between the two arms of the U-shaped core across the member 156, to form an evenly distributed annular field if viewed in the axial direction. The polarity can be changed by changing the current direction in the coil. The annular permanent magnet 152 has a similar U-shaped cross-section with a non-magnetic member 153 in it to separate the two poles S and N of the magnet. Obviously, the two magnets 152 and 154 will produce between them a repulsive or attractive force according to the current direction in the coil 155. These two magnets are arranged to have their opposing surfaces in generally conical shape (similar to that of the bearing member 52 shown in Fig. 2) which are complementary to each other so that a centring force formed between them would tend to keep them in a coaxial relationship. The non-magnetic members 153 and 156 are preferably made of low friction materials, such copper, bearing steel or Teflon, so that they can have between them low friction sliding contact as a further buffer arrangement to the rotor's axial movement.

The operation of this suspension mechanism is as follows.

Before the stator 120 is switched on, each of the electromagnet 154 is supplied with a current of the same value, to generate from two ends of the rotor two opposite repelling (or attracting) forces which are of the same strength. These opposite forces would "clamp" the rotor at its neutral and balanced position and at the same time keep it radially suspended in the pump body. When an operating current is supplied to the stator, which drives the suspended rotor to rotate, a control signal which is proportional to the stator's average operating current is supplied to each of the electromagnets 154 with the effects of increasing the repelling force of the one at the upstream end of the rotor (left-hand end in Fig. 3) while reducing the repelling force at the other end. The combined effect by this control signal is a net force directing towards the downstream end of the rotor, which is sensitive to the rotor's movement and tends to balance the thrust force on the rotor produced by the axial flow of the liquid. In this way, the axial position of the rotor is automatically maintained. When the rotational direction of the rotor is reversed by changing the direction of the stator's operating current, so is the direction of the control signal supplied to the electromagnets 154, with the effects of having a reversed balancing force automatically.

In order to make further accurate adjustment of this sensitive suspension bearing, sensors 166 are fitted to each end of the pump body which constantly monitor the axial position of the rotor. The output signals from the sensors are sent to a control unit (not shown) to further adjust the current supplied to the electromagnets 154, so that when a significant displacement of the rotor is detected, the bearings 150 can be further adjusted, e.g. having one of them generating an repelling force while the other an attracting force to help to return the rotor to its neutral position. The sensors 166 can be any convention displacement sensor, such as capacitive sensors, Hall's effect sensors, infrared sensors or ultrasonic sensors, all of them are known in the art.

Thirdly, the spiral vanes 149 formed on the outer surface of the rotor 140 work in a way similar to that of the first embodiment, i.e. producing a peripheral fluid cushion which operates as a hydraulic bearing. This hydraulic bearing is formed by sucking water into the magnetic gap 172 from the downstream end of the rotor, as shown by arrows <u>In</u> at the right-hand end of the rotor 140 in Fig. 3, driving it to flow backwards and returning it to the mainstream flow at the upstream end of the rotor, as shown by the arrows <u>Out</u>. By

arranging a throttle ring 160 at each end of the rotor 140, this hydraulic bearing also works to damp any axial oscillation of the rotor. For example, when the rotor experiences a sudden increase of the thrusting force, e.g. due to a sudden increase of the output resistance or the increase of the input electric power, it would be forced to move backwards to the upstream end. This brings the throttle ring 160 at the upstream end closer to the member 154 shown in Fig. 4, therefore at least partially block the peripheral gap between the member 154 and the tubular member 142 which serves as an outlet for the back flow circulation. The reduced outlet will at the same time lead to an increase of the water pressure in the space between the magnets 152 and 154 which is a part of the gap 172, that in turn resists the rotor's further axial movement. It should be noted that this damping effect works when there is a sudden movement of the rotor so it protects the free rotor from hitting the pump body. The gradual change of the rotor's axial position is mainly counter-balanced by the forces provided by the magnetic registration and the suspension bearings 150 as explained above.

In Fig. 3, it is also shown that the tubular member 142 has an increased thickness compared with that of the first embodiment. This is made to form a buoyant structure by using light materials, such as plastics or resin, to make the number 142, or by using metal material with channels or cavities filled with light material to reduce its total weight. The intended effects are to make the rotor 140 as a whole to have a gravity close to the liquid to be pumped, so that when the pump is filled with the liquid, the rotor "floats" in the liquid. This will help to suspend the rotor in the liquid and reduce the radial and axial oscillation of the rotor relative to the stator when the rotor is fully suspended by the magnetic and hydraulic forces. It should be mentioned that the portion 143 at each end of the tubular member 142 forms a narrowed entrance for liquid to enter the inside space of the rotor, which has the effect of increasing the flow speed at each end of the rotor. This is intended to reduce the opportunity for the solid particles carried by the flow, such as sand or rust, to enter the gap 172 between the rotor and the stator. The operation of this arrangement is described as follows.

When the solid particles are carried by the flow to the inlet end of the rotor 140 (the left-hand end in Fig. 3), they are driven to the centre of the flow by the back flow coming out of the gap 172, as shown by the arrows <u>Out</u>. This annular outlet of the back flow joins the mainstream and further increase the flow speed at the narrow entrance defined by the portion 143, which helps to carry the solid particle, if any, into the hollow rotor. Once the particles

are inside the rotor, because they are heavier than the liquid, they would be urged by the centrifugal force of the rotor against the inner surface of the fluid passage and be moved forwards by the axial flow. As they are moved to the outlet end, the shape of the portion 143, which accelerates the flow and the particles at the outlet, helps to "shot" the particles out of the downstream end of the rotor, along the line shown by the arrow S, before they have a chance to move into the gap 172. Only clean liquid will be sucked into the gap 172, as shown by the arrows <u>In.</u>

Fig. 4 shows a third preferred embodiment of the present invention, which is actually the second embodiment adapted to be used in upright position. Therefore, the reference numerals in this figure are same as those in Fig. 3 for those parts which are not changed. As shown in Fig. 4, the general structure and the operation principles of the pump 100 are similar to that of the previous embodiments, therefore only the different features of this embodiment are to be described hereinbelow.

When the pump 100 is used in this upright position, the need for preventing the solid particles from entering the gap 172 is not so important and the narrow entrance formed by the portion 143 is not necessary. In this case, the suspended rotor 140 is under a much greater counter force for it has to support the weight of the whole column of liquid above it. In order to compensate the weight of the liquid on the rotor 140, an additional magnetic bearing system 190 is fitted to the lower end of the pump, which includes an electromagnet formed by an annular magnetic core 194 and a coil 195 carried by the cap 130, and a permanent magnet 192 carried by the rotor 140. As shown in Fig. 4, the magnet 192 is arranged not in perfect axial registration relative to the electromagnet 194, so that when the electromagnet is energized the magnet 192 hence the rotor 140 is subject to an upward magnetic force which counter-balance the weight of the liquid on the rotor. Only one such axial bearing system is shown in Fig. 4, but obviously more can be fitted to one or both ends of the rotor to produce enough balancing force.

The arrangement of Fig. 4 can also be used as a hydraulic electric generating system when water is allowed to flow down to drive the rotor to rotate, then the operation of the magnets 144 and the coil 126 would be reversed to generate electricity. When the system is made small and light, it can also be used as a flow-meter which produces an electric signal in proportion with the flow rate of the liquid passing through it.

Figs. 5A to 7 show a pump 200 according to a fourth preferred embodiment of the present invention. Many features in this embodiment are similar to those of the previous embodiments so only the new features are fully described here.

As shown in Fig. 5A, the pump 200 has a housing 220 and two cap members 230, each carries a bearing electromagnet 250, all similar to those shown in Fig. 3. Inside the housing 220, there are two stators 210 and 210', separated by an annular separator 221 which carries a central bearing member 240, to be fully described below with reference to Fig. 7. Within the cylindrical inner space defined by the two stators there is a rotor assembly formed by two hollow rotors 260 and 260' each matching a corresponding stator, an annular connector 270 for connecting the two rotors to each other, and a driving mechanism 280 fitted inside the hollow structure of the rotor assembly. Electric connections (not shown) are made to the two stators so that they produce same rotary electromagnetic fields to drive the two rotors to rotate together. A perspective view of the rotor assembly is shown in Fig. 6 with the details of the connection between the connector 270 and the two rotors shown in Fig. 7. The general structure and bearing arrangements for each rotor 260 or 260' are similar to that shown in Fig. 3, therefore their description does not need to be repeated.

The new driving mechanism 280 includes two securing rings 281 and 281' for fasting the mechanism to the rotor assembly, each at one end thereof; a lining tube 282 being clamped between the two rings; a number of screw blades 283 of flexible and elastic material being fitted inside the tube 282; and tow spiral wire springs 284 and 286 being fitted around the outer surface of the tube 282. The blades 283 are made to have an outer diameter slightly larger than the inner diameter of the tube 282 so that when the blades are fitted into the tube 282 they tend to expand, forming a tight fit therebetween. Each blade 283 has a number of studs 288 projecting radially outwards from its outer edge and being evenly located along the blade's length, which studs are fitted into an array of corresponding slots 287 formed in the wall of the tube 282 to keep the blades physically engaged with the tube 282 and to be rotatable therewith.

The two springs 284 and 286 outside the tube 282 are arranged in opposite spiral senses, with the spring 284 in the same spiral manner as the blades 283, and each spring has one end secured to an end of the tube with the other end joined to a connecting ring 285, which is also slidably fitted around the tube 282. Such an arrangement allows smooth sliding

movements of the springs and the blades, as to be described later. The studs projecting from the blades are engaged by the springs so the blades are biased by the springs. Under the conditions as shown in Fig. 5, the spring 284 tends to expand along its axial direction while the spring 286 tends to contract, thus they produce a joint biasing force on the blades to urge them towards the downstream end of the pump (i.e. the right-hand side in Fig. 5A) and keep them there in their fully expanded status.

The shape and orientation of the slots 287 are different from one another and the array of slots are arranged in a way that their lengths are progressively increased along the flow direction shown by the arrow A. That is to say that the downstream end of each blade is relatively "free" because its stud is fitted in a long slot which gives it more room to move, compared with the stud at the upstream end, which is virtually fixed. The purposes for this arrangement is to make the blades 283 compressible during the pump's operation by allowing the studs 288 to slid within and along the slots 287, so as to keep the blades always engaged with the tube 282. It is to be noted that when the blades 283 are compressed, they tend to expand radially but this radial expansion is restricted by the tube 282. The result is for the blades to twist while being squeezed between it two ends. The shape and length of each slot is therefore made to accommodate this twisting factor to ensure blades' smooth movement during their compression.

In operation, when the rotating rotor assembly produces a forward fluid driving force, the fluid would produce a backward thrust to compress the blades. In a low load operation, this counter force is balanced by the elasticity of the blades and the combined biasing force of the two springs 284 and 286. However, when the counter thrust is increased beyond a limit, for example when there is a big increase of the output resistance or a big increase of input driving power, the counter thrust on the blades would overcome the biasing forces and cause the blades to be compressed backwards. Once this axial movement happens it would increase the biasing forces provided by the springs so the balance would quickly be established at a new position, where since the blades are compressed to a smaller pitch (see P2 in Fig. 5B in contrast with P1 in Fig. 5A), the pump operation would be stabilised again for a smaller flow rate under a higher output pressure, without affecting the rotor's rotating speed. That is to say the pump with such an adjustable driving mechanism can automatically and instantly response to changes of output resistance or input power, or both, from an operation of low pressure and high flow rate to one of high pressure and low flow rate, or

vise versa, without compromising its energy efficiency. This makes the pump operable over a much wider range of working conditions and capable to provide smooth operation during its start-up or slow-down. This adjustability is particularly desirable when the pump is used as a marine propeller for, e.g. a speedboat, where quick acceleration/deceleration and smooth transformation under changing conditions are essential for good performance.

Now referring to Fig. 7, in which the driving mechanism 280 is removed to show a cross-sectional and partially exposed view of the details of the central bearing member 240 and the annular connector 270. The combination of the central bearing member 240 and the annular connector member 270, together with the magnets 264 and 264', provide both magnetic and hydraulic suspension bearing effects. For the sake of easy understanding, these two aspects are to be explained separately.

Firstly, the magnetic suspension bearing is described. The annular connector 270 has a base ring 271 made of a magnetic material, such as iron, which connects the two rotors 260 and 260' to each other by snap engagements 274, so as to form an integrated rotor assembly. The base ring 271 also serves as a magnetic bridge for connecting the two magnets 264 and 264' to form a complete magnetic ring with corresponding poles matching that of the two annular poles of the core member 241 of the electromagnet 240, separated by a non-magnetic member 243. In operation, when the coil 242 is energised, the electromagnet 240 and the permanent magnets 264 and 264' form a suspension pair with their corresponding poles opposing one other, causing mutual-repelling, so as to "lock" the rotor assembly to its neutral position as described above.

Secondly, the hydraulic suspension is explained. When a liquid is filled into the pump structure, it enters the gap between the tube 282 and the inner surface of the rotor assembly via the slots 287. The base ring 271 has several rows of small holes 273, also shown in Fig. 6, which are covered by a filter member 272 which soaks up the liquid once it enters the gap. When the rotor assembly starts to rotate, liquid soaked up by the member 272 will be spun off by centrifugal forces through the holes 273 to form a tangential flow indicated by the arrows S in Fig. 6. This tangential flow of the spin-off liquid is equally divided by a dividing edge 244 on the inner surface of the dividing member 243 to form two separated flows, each of a width "d", and to be directed by the guiding fins 245 or 245' towards the correspond spiral vanes 269 or 269' to form two opposite bearing flows represented by the arrows B.

The remaining parts of each bearing flow passage are similar to that of Fig. 3. Since these two bearing flows produce equal-sized but oppositely directed bearing forces, they make further contributions in keeping the rotor assembly close to its neutral position. It should be mentioned that the width "d" to each side of the dividing edge 244 is about the same as the width of the axial registration members 217 and 267, for purposes to be described below.

In operation, the rotating rotor assembly is always kept within a small axial range of "d" to each side of its neutral position although there is no physical support except the suspension bearing arrangements of the present invention. Whenever the rotor assembly is forced to move away from its neutral position, e.g. due to the counter thrust caused by the axial flow, the base ring 271 would also move relative to the fixed dividing edge 244 so that the holes to the one side of the dividing edge 244 would be increased with a corresponding decrease to the other side. This change will cause an imbalance between the two bearing flows, which is enhanced due to the effects of the throttle ring 268 or 268' at the downstream end of each bearing flow B. Because of this shift of balance, any further movement of the rotor assembly form its neutral position would meet increased resistance until the rotor stabilised at a new balanced position. Furthermore, when the rotor assembly is moved away from it neutral position for a distance close to the size of "d", the misalignment between the rings 217 and 267 would be at its maximum, causing a maximum registration force to return the rotor assembly to its neutral position, further restricting the rotor assembly's deviation from its neutral position. Finally, it should be mentioned that additional suspension control can be conducted according to the sensing signals provided by the sensors (166 in Fig. 3) fitted in each cap members, as described before.

The assembly of the pump 200 can be made in the following procedure. First of all, the electromagnet 240 and the separator 221 are made as one member by moulding the separator 221 around the electromagnet and forming the member 243 at the same time. The two stators 210 and 210' are then fitted to each side of the moulded member, and the three of them are inserted into the housing 220 to form the tubular stator structure. After this is done, the connector 270 is located in the inner space of the stator structure and the two rotors 260 and 260' are snap-fitted to the connector from two ends to form the rotor assembly. Once the rotor assembly is fixed, the driving mechanism 280 can be assembled by first fitting the end ring 281' to one end of the rotor assembly, then sliding the tube 282 together with the springs and the blades into the rotor assembly from the other end. The

tube 282 engages the ring 281' by the projection 289' at its leading end and also engages with the channels formed on the inner surface of the rotor 260 by the ribs indicated by the dash lines 289 at the other end, so that the tube 282 is carried to rotate by the rotor assembly. Then the end ring 281 is fasted to the other end of the rotor assembly. Finally the two cap members 230 are fitted to the housing to produce a finished pump.

Figs. 8 to 8D show a pump 300 according to a fifth preferred embodiment of the present invention. Among them, Fig. 8 is a cross-sectional view of a pump 300; Fig. 8A is an enlarged view of a bearing mechanism 380 shown in Fig. 8; Fig. 8B is a cross-sectional view of the bearing mechanism 380 taken along the line B-B in Fig. 8A; Fig. 8C is a cross-sectional view of an impeller 350 shown in Fig. 8; and Fig. 8D is a cross-sectional view of the impeller 350 taken along the line D-D in Fig. 8C. Many features in this embodiment are similar to those of the previous embodiments so only the new features are to be described.

As shown in Fig. 8, the pump 300 is to be kept upright. It has a housing 320, a top cap member 330 and a base member 340. Inside the housing 320, there is a stator 310 which has axial registration rings 317, as described with reference to Fig. 3, and a rotor assembly formed by a hollow rotor 360, an impeller member 350 connected to the bottom end of the rotor 360 and a pivot bearing mechanism 380 fitted inside the rotor close to its upper end. A supporting shaft 370, which is not in section, is also fitted inside the rotor extending between the base 340 and the bearing 380.

More particularly, the housing 320 has a cylindrical portion defining an upper motor chamber for the stator 310 and the rotor 360, and an extended lower portion 326 defining, together with the base member 340, an impeller chamber for the impeller 350. An annular part 324 of the housing 320 separates the two chambers and forms a support for the stator 310. Liquid inlet holes 328 are formed around the lower part of the portion 326, which holes are covered by a filter 391. The base member 340 has a central portion 342 of a generally conical shape protruding into the impeller chamber to define with the housing portion 326 an annular and generally conical inner space for the similarly shaped impeller 350. The base 340 also has adjustable supports 349 for keeping the pump upright. Liquid inlet holes 344 are formed in the base member and covered by a second filter 392. The purpose of using the filters 391 and 392 is to prevent any solid particles from entering the interior of the pump which may cause damages to hydraulic bearing vanes and surfaces.

They are not incorporated when the pump is used for pumping clean liquids or highly viscous or pasty stuff. At the centre of the base portion 342 there is a cylindrical protrusion which has a central hole 346 of a polyhedral cross-section for receiving the low end of the shaft 370, to be described later. A throttle ring 348 is fitted around the cylindrical protrusion for hydraulic bearing effects, also to be described later. The cap 330 is similar to that of the previous embodiments, and it has a top portion with a screw thread 332 for connecting to a pipeline or hose.

The rotor 360, similar to that shown in Fig. 4, has magnets 364 and axial registration rings 367 matching that of the stator 310, inner screw blades 366 for driving the liquid flow, outer spiral vanes 369 for hydraulic bearing and a throttle ring 368 fitted at its top end. Inside the hollow space of the rotor 360 is fitted the supporting shaft 370 which has screw blades 372 in a spiral direction opposite to that of the blades 366. The lower end of the shaft 370 has a polyhedral connector which is snap-fitted in the polyhedral hole 346 of the base member 340 to ensure that the shaft is not rotatable. The top end of the shaft is fitted with a bearing base 376, made of a very hard material for forming a gimbal mount with the tip of the pivot bearing 380.

As shown in Figs. 8A and 8B, the bearing mechanism 380 includes a ring member 381 with a number of engaging teeth 384 for snap-engaging with the end portion 362 of the rotor 360. The teeth 384 ensure that the ring 381 is fixed to the rotor and the mechanism 380 as a whole would rotate with the rotor 360 during pump operation. A central part 383 is connected to the ring 381 by a number of connecting members 382 which also serve as propelling blades when they rotate with the rotor. The central part 383 has a hexahedral hole for accommodating a pin 385 of the same cross section, so that the pin rotates with the rotor. The pin 385 is secured by a nut 387 at its upper end and biased downwards by a spring 388 at its lower end. The tip 386 at the lower end of the pin 385 is made of a very hard material, such as ceramic, glass or super-hard metal, for forming a single point bearing contact with the bearing base 376 of a similarly hard material. The use of the spring 388 ensures that the tip 386 is always biased against the base 376, so that although the rotor assembly as a whole may have a small degree of axial movement, of a distance "d" as shown in Fig. 8, it is prevented from any direct impact with the hard bearing base due to sudden changes of output load or electric driving input. It should be noted that when in operation, the whole rotor assembly formed by the members 380, 360 and 350 is borne, except the effects of the

hydraulic bearings and the axial registration arrangement, solely by the tip 386 on the base 376 which provides a gimbal mount of high stability and low wear and rotating resistance, and no magnetic bearing is used in this embodiment.

Figs. 8C and 8D show the details of the impeller 350 which is to be secured to the lower end of the rotor 360 by an engaging ring 358, so as to rotate with the rotor. The engaging ring itself can be made as a separate member which is then secured to the impeller by a similar engaging arrangement. The engagement can be of the similar construction as that of the ring 381 shown in Fig. 8A. The impeller 350 has a hollow conical body 352 with outer blades 354 and inner vanes 356. The blades and vanes are in the same spiral direction so that when the impeller rotates with the rotor, the blades 354 produce a generally upward liquid flow from the holes 328 towards the hollow rotor while at the same time the vanes 356 produce an upward bearing flow which keeps the impeller "floated" on the outer surface of the base portion 342. The bearing flow is eventually forced to pass over the throttle ring 348 and enters the hollow rotor via the annular gap between the inner surface of the cylindrical portion 357 and the outer surface of the central protrusion of the base member 340. That is to say, when the rotor assembly rotates at a stable speed, the hydraulic bearing flow formed by the vanes 369 on the outer surface of the rotor 360 and that by the vanes 356 on the inner surface of the body 352 would provide joint bearing effects which keep the whole rotor/impeller assembly "floated" and also lubricated, therefore significantly reduce the bearing load on the pivot tip 386.

In operation, once the rotor 360 starts to rotate under the influence of the magnetic driving force produced by the stator 310, it would carry the impeller 350 and the bearing mechanism 380 to rotate with it. The liquid sucked into the impeller chamber via the holes 328 and 344 would be forced to flow upwards by the blades 354 and the vanes 356 to enter the hollow rotor, where it would be forced to flow upwards by the effects of the blades 366 which are rotating and the blades 372 on the supporting shaft 370, which are not rotating. Since the blades 372 are in a spiral direction opposite to that of the blades 366, they work together to reduce the swirling factor of the liquid flow passing between them and to increase the upward driving force, therefore producing a significantly increased output pressure. This upward flow is further promoted by the effects of the blades 359 of the impeller 350 and the blades 382 of the bearing 380, both sets rotate with the rotor.

It is easy to understand that when there is a sudden change of load or electric input, the rotor assembly would tend to move axially. This tendency is compensated by the elastic bearing force by the gimbal bearing 380 and the two throttle rings 348 and 368. For example, assuming there is a sudden drop of output pressure due to the fact the hose connected to the top cap 330 is burst under the output pressure, the whole rotor assembly would unavoidably move upwards, which leads to the situation that the upper throttle ring 368 would block the bearing flow inlet to the hydraulic bearing on the outer surface of the rotor 360, and at the same time the gap defined by the lower throttle ring 348 is increased so much that the vanes 356 on the inner surface of the impeller 350 would not be able to produce bearing effects. That is to say both hydraulic bearing arrangements stop to provide upward bearing forces so the whole rotor assembly would move to a new balanced position. Obviously, when the rotor assembly moves downwards, e.g. due to an increase of the output pressure, the arrangement works in the opposite way to balance the system automatically.

It is also easy to understand that since the pump 300 has two sets of liquid inlet holes 328 and 344, it can be conveniently used as a mixing pump with the holes 328 for the main liquid component while the holes 344 for adding a second component which would be fully mixed with the main flow in the hollow rotor. This is also useful in the case where the main flow is a highly viscous or thick mixture and a lubricant and/or diluent liquid can be introduced via the holes 344 to keep the system lubricated to reduce flow resistance. All the structural components of this pump, except the electric or magnetic parts and the hard gimbal bearing tip and base, can be made by moulding plastics or resin, therefore it is easy to achieve high precision and low manufacturing costs.

Fig. 9 shows a cross-sectional view of a pump 400 according to a sixth embodiment of the present invention. The pump 400 has a housing 420 accommodating two stators 410 and 410' separated by a separator 421, and two rotors 460 and 460', which are kept between two cap members 430 and 430', each carrying a gimbal bearing 480 or 480'. Generally speaking, the stator/rotor combination 410 and 460 work in a way similar to that of Fig. 4, with hydraulic and magnetic bearings and an magnetic axial registration arrangement formed between them. The operation of the stator/rotor combination 410' and 460' is similar.

The new features of the rotor 460' include the central propelling member 470' which is integral with the screw blades 466' and the hollow rotor body. The member 470' has a

polyhedral central hole. The hole engages at its upper end the connector part 474 of a central propelling shaft 470, which is similar to the central shaft 370 shown in Fig 8. The low end of the hollow member 470' engages the bearing base of the gimbal bearing 480' supported by the cap 430', while the upper end of the propelling shaft 470 carries the bearing base of the other gimbal bearing 480 fitted in the cap 430. That is to say, the two bearings 480 and 480' "clamp" between them the combination of the propelling shaft 470 and the whole rotor 460', which are made to rotate together. The structure of the bearings 480 and 480' is similar to that of the bearing 380 shown in Figs 8A and 8B, except that the connecting members 382 are arranged along the axial direction because they do not work as propelling blades.

When in operation, the rotor 460 rotates in direction shown by the arrow sign "R" on top of the drawing, which produces an axial flow in direction "A". On the other hand, the rotor 460' and the shaft 470 rotate in an opposite direction shown by the arrow sign "r" at the bottom, and produce an axial flow in direction "a". Since the spiral direction of the blade 466 is opposite to that of the blades of the shaft 470, and they rotate at the opposite directions, this arrangement ensures that the two sets of screw blades always cancel each other's swirling effects to the liquid flow and at the same time enhance the common axial direction driving force. That is to say, a significantly increased output pressure is achieved without sacrificing the flow rate.

The rotor 460' can be manufactured by first making a central propeller 470' with integrated screw blades 466'. The hollow rotor 460' is made as a separate part with spiral channels formed on its inner surface, matching the outer edge of the blades 466'. Then the propeller 470' is fitted into the rotor 460' by screwing the blades in the channels. Adhesive is used to bind the blades 466' to the rotor body. Since these parts can be easily made of plastics or resins, the manufacturing costs are very low.

The pump 400 is suitable for forming a high pressure fluid jet of a stable flow rate. In case of high pressure applications, a number of them can be connected by simply screw one to another. Because of its very compact structure, it can be conveniently fitted to the outlet end of a pipeline. That means the whole pipeline can be operated under a much lower internal pressure for transferring the fluid medium, therefore the tremendous costs and

difficulties of using and maintaining high pressure pipes and associated connectors are avoided without compromising the operational requirements.

Industrial Applicability

From the above description it is clear that the pump/propeller of the present invention can be easily produced because the stator, the caps and the bearings systems can be made by conventional methods. The simple and highly symmetrical structures disclosed in the present application make them suitable for high quantity mass-production and high level quality control, therefore very low cost per item. The caps and the housing should be made of non-magnetic materials, such as aluminium, stainless steel, copper, plastics or fibre reinforced resin. Similarly, the tubular member of the rotor together with the propelling blade(s) should also be made of non-magnetic material. The screw blade(s) can be made as a separate member first then welded to a tube or casting/moulding the tube around it. Alternatively, the whole structure can be formed by casting around a cylinder die with a spiral channel for forming the blade(s), then the final product can be unscrewed from the die. A continuous casting process can also be used to from a long tube with an internal blade, then it can be cut to required length for making the tubular rotor member. The number of the blades, its height h, pitch p and propelling angle α, as shown in Fig. 1, can be changed or adjusted to suit the needs of a particular application.

The internal blade structure of the rotor provides a clog-free propelling structure which is able to propel through it anything that can by all means enter the pipeline at the first place, therefore it is suitable to a wide variety of applications, especially when used for handling liquids carrying a high proportion of solid contents, e.g. applications as marine propeller, sludge pump, sewerage pump or pump for impelling or injecting pasty or viscous mixtures. The relatively long screw or spiral blade is able to distribute stress evenly over its whole length so it can undergo large load without causing over-stress. Because the internal structure of the pump can be wholly sealed, it is also suitable for submerged applications and/or leakage-free applications. Furthermore, it is easy to fit such a pump into an existing pipeline at any desirable position since the pump can be configured in roughly the same dimension as a piece of tube, and a number of pumps can be connected in series into the same pipeline to increase the total driving force, or to have one or more of them kept idle as

back-up units. When an idle pump is fitted to a pipeline, its rotor will be able to rotate freely therefore not causing significant resistance to the flow through it. The hydraulic bearing vanes can also be formed on the inner surface of the stator, instead of or in addition to that formed on the outer surface of the rotor. However it is easier to form them on the outer surface of the rotor from the point of view of machine operation. In the above description the term "spiral vane" should be interpreted to include both the form of projecting ribs, as shown in Figs. 1, 3 and 4, and spiral grooves or channels, which are easier to manufacture and can produce the satisfactory effects, especially when elastic materials are used for forming them. It is worth mentioning that in the accompanying drawings the size of the bearing vanes is exaggerated for the sake of easy recognition. In practice they are made very small to ensure a close fit between the moving parts.

It should be understood that many of the features described above can be combined with one another in different manners, to suit the needs of different applications of the present invention. For example, when the pump is used for pumping a clean liquid, the arrangement for preventing solid particles from entering the magnetic gap of the motor or the filters would not be necessary. On the other hand, if the magnetic registration and/or suspension mechanism can maintain a satisfactory suspension control, the hydraulic back-flow can be changed to a forward flow to further increase the operation efficiency. Furthermore, in a relatively small pump the magnetic suspension bearing 150 can be formed by two permanent magnets arranged to repel each other, instead of using an electromagnet 154. This will reduce electric power consumption and the total weight of the pump. Other modifications and alternations can also be done by those skilled in the art without departing from the concept and the principles of the present invention.

Claims

1. An axial flow pump/marine propeller comprising:

a pump body with two open ends;

a stator member carried by the body and defining a cylindrical inner space extending between said two open ends of the pump body;

a tubular rotor member with means for propelling fluid therethrough when the rotor member rotates being rotatably fitted in the inner space; and

electromagnetic means arranged between the stator and the rotor for generating a rotational magnetic field for driving the latter to rotate;

wherein the rotor member is supported by a suspension bearing mechanism which provides rotational and thrust bearing for retaining the rotor member at a balanced position in response to its rotational and/or axial movements.

2. An axial flow pump/marine propeller according to claim 1, wherein the rotor member and/or the stator member has spiral bearing means formed on at least one of the surfaces defining the gap between said two members, for forming a peripheral flow therein when the rotor member rotates.

3. An axial flow pump/marine propeller according to claim 2, wherein said spiral bearing means is formed by an elastic material.

4. An axial flow pump/marine propeller according to any of the preceding claims, further including two mechanical bearings, each fitted to an end of the rotor.

5. An axial flow pump/marine propeller according to any of the preceding claims, further comprising sealing means fitted to each end of the rotor so as to seal the gap between the stator and the rotor, so that a lubricating liquid can be filled in the sealed gap.

6. An axial flow pump/marine propeller according to claim 5, wherein a fluid circuit is formed between the stator and the rotor so that a cooling fluid can be circulated therein.

7. An axial flow pump/marine propeller according to claim 5 or claim 6, further comprising means for supplying under a stable pressure said lubricating liquid and/or cooling fluid into said sealed gap.

8. An axial flow pump/marine propeller according to any of the preceding claims, wherein said electromagnetic means include annular members of high magnetic resistance fitted to the corresponding positions on the inner surface of the stator and the outer surface of the rotor, so as to form a magnetic registration mechanism therebetween to keep the stator and the rotor in axial alignment.

9. An axial flow pump/marine propeller according to any of the preceding claims, wherein said bearing means include at least one magnetic suspension bearing arrangement comprising: an annular electromagnet member with tow annular magnetic poles and an annular permanent magnet member with two magnetic poles matching that of the electromagnet member so that an repelling or an attracting force can be selectively generated therebetween by controlling the electric current to the electromagnet member.

10. An axial flow pump/marine propeller according to claim 9, wherein the annular magnetic poles of the electromagnet and permanent magnet members are arranged to form complementary conical surfaces so as to provide both rotary and thrust suspension.

11. An axial flow pump/marine propeller according to claim 9, wherein the annular magnetic poles of the electromagnet member and the permanent magnet member are arranged to form complementary cylindrical surfaces so as to provide thrust suspension.

12. An axial flow pump/marine propeller according to claim 10 or claim 11, further comprising a control unit and sensing means for adjusting the electric currents supplied to the suspension bearings to maintain the axial position of the rotor according to the current supplied to the stator and/or the signals indicating the axial position of the rotor.

13. An axial flow pump/marine propeller according to any of the preceding claims, wherein the rotor as a whole is made to have a gravity close to that of a liquid to be propelled, so that the rotor is floated when immersed in the liquid.

14. An axial flow pump/marine propeller according to claim 2, wherein said gap is in fluid communication with said fluid passage and said spiral bearing means is arranged so that the peripheral flow is formed to circulate between the gap and the fluid passage.

15. An axial flow pump/marine propeller according to claim 14, wherein a throttle ring is fitted to an axial end of said gap, which ring restricts the peripheral flow when the rotor has an axial movement towards that end.

16. An axial flow pump/marine propeller according to any of the preceding claims, wherein the hollow structure of the rotor has a narrowed portion to increase the flow speed at that position.

17. An axial flow pump/marine propeller according to any of the preceding claims, wherein the suspension bearing mechanism includes a fluid suspension system with flow dividing means for forming two opposite peripheral flows, so as to keep the position of the rotor member balanced during its rotational operation.

18. An axial flow pump/marine propeller according to claim 17, wherein said flow dividing means further comprises means for adjusting said two opposite peripheral flows in response to the axial movement of the rotor member, to keep the same self-balanced.

19. An axial flow pump/marine propeller according to any of the preceding claims, wherein said means for propelling the fluid includes one or more flexible spiral blades which are compressible in axial direction in response to the change of power input or working load.

20. An axial flow pump/marine propeller according to claim 19, wherein said flexible spiral blades are biased by elastic means to keep them axially expanded.

21. An axial flow pump/marine propeller according to any of the preceding claims, wherein said propelling means includes two co-axially arranged driving members with spiral means in opposite spiral directions so that when they have relative rotational movements, they cancel each other's swirling effects to produce a high pressure axial output.

22. An axial flow pump/marine propeller according to claim 21, further comprising a set of a second stator member, a second rotor member and second electromagnetic means so that said two driving members are driven to rotate in opposite directions.

23. An axial flow pump/marine propeller according to claim 21 or claim 22, wherein the central one of said two driving members is supported by a pivot bearing arrangement.

24. An axial flow pump/marine propeller according to any of the preceding claims, wherein said means for propelling the fluid includes a conical impeller member fitted to the upstream end of the pump to further promote the axial flow.

25. An electric generator/flow-meter which has a structure as specified in any of the preceding claims.

26. An axial flow pump/marine propeller and/or an electric generator/flow meter constructed substantially as described herein with reference to Figs. 1 and 2, Fig. 3, Fig. 4, Figs. 5A to 7, Figs. 8 to 8D or Fig. 9 of the accompanying drawings.

Amendments to the claims have been filed as follows

1. An axial flow pump/marine propeller comprising:

a pump body with two open ends;

a stator member carried by the body and defining a cylindrical inner space extending between said two open ends of the pump body;

a tubular rotor member in the inner space for defining a fluid passage with means for propelling fluid therethrough; and

electromagnetic means arranged between the stator and the rotor for generating a rotational magnetic field for driving the latter to rotate;

wherein the rotor member is supported by a suspension bearing mechanism which has self-balancing means for stabilising the rotor member's suspended rotation by providing thereto thrust compensation in response to changes in its rotational movements and/or axial position.

2. An axial flow pump/marine propeller according to claim 1, wherein the rotor member and/or the stator member has helical bearing means formed on at least one of the surfaces defining the gap between said two members, for forming a peripheral flow therein when the rotor member rotates.

3. An axial flow pump/marine propeller according to claim 2, wherein said helical bearing means is formed by an elastic material.

4. An axial flow pump/marine propeller according to any of the preceding claims. further including two mechanical bearings, each fitted to an end of the rotor.

5. An axial flow pump/marine propeller according to any of the preceding claims. further comprising sealing means fitted to each end of the rotor so as to seal the gap between the stator and the rotor, so that a lubricating liquid can be filled in the sealed gap.

6. An axial flow pump/marine propeller according to claim 5, wherein a fluid circuit is formed between the stator and the rotor so that a cooling fluid can be circulated therein.

7. An axial flow pump/marine propeller according to claim 5 or claim 6, further comprising means for supplying under a stable pressure said lubricating liquid and/or cooling fluid into said sealed gap.

8. An axial flow pump/marine propeller according to any of the preceding claims, wherein said electromagnetic means include annular members of high magnetic resistance fitted to the corresponding positions on the inner surface of the stator and the outer surface of the rotor, so as to form a magnetic registration mechanism therebetween to keep the stator and the rotor in axial alignment.

9. An axial flow pump/marine propeller according to any of the preceding claims, wherein said bearing means include at least one magnetic suspension bearing arrangement comprising: an annular electromagnet member with tow annular magnetic poles and an annular permanent magnet member with two magnetic poles matching that of the electromagnet member so that an repelling or an attracting force can be selectively generated therebetween by controlling the electric current to the electromagnet member.

10. An axial flow pump/marine propeller according to claim 9, wherein the annular magnetic poles of the electromagnet and permanent magnet members are arranged to form complementary conical surfaces so as to provide both rotary and thrust suspension.

11. An axial flow pump/marine propeller according to claim 9, wherein the annular magnetic poles of the electromagnet member and the permanent magnet member are arranged to form complementary cylindrical surfaces so as to provide thrust suspension.

12. An axial flow pump/marine propeller according to claim 10 or claim 11, further comprising a control unit and sensing means for adjusting the electric currents supplied to the suspension bearings to maintain the axial position of the rotor according to the current supplied to the stator and/or the signals indicating the axial position of the rotor.

13. An axial flow pump/marine propeller according to any of the preceding claims, wherein the rotor as a whole is made to have a gravity close to that of a liquid to be propelled, so that the rotor is floated when immersed in the liquid.

14. An axial flow pump/marine propeller according to claim 2, wherein said gap is in fluid communication with said fluid passage and said spiral bearing means is arranged so that the peripheral flow is formed to circulate between the gap and the fluid passage.

15. An axial flow pump/marine propeller according to claim 14, wherein a throttle ring is fitted to an axial end of said gap, which ring restricts the peripheral flow when the rotor has an axial movement towards that end.

16. An axial flow pump/marine propeller according to any of the preceding claims, wherein the hollow structure of the rotor has a narrowed portion to increase the flow speed at that position.

17. An axial flow pump/marine propeller according to any of the preceding claims, wherein the suspension bearing mechanism includes a fluid suspension system with flow dividing means for forming two opposite peripheral flows, so as to keep the position of the rotor member balanced during its rotational operation.

18. An axial flow pump/marine propeller according to claim 17, wherein said flow dividing means further comprises means for adjusting said two opposite peripheral flows in response to the axial movement of the rotor member, to keep the same self-balanced.

19. An axial flow pump/marine propeller according to any of the preceding claims, wherein said means for propelling the fluid includes one or more flexible spiral blades which are compressible in axial direction in response to the change of power input or working load.

20. An axial flow pump/marine propeller according to claim 19, wherein said flexible spiral blades are biased by elastic means to keep them axially expanded.

21. An axial flow pump/marine propeller according to any of the preceding claims, wherein said propelling means includes two co-axially arranged driving members with spiral means in opposite spiral directions so that when they have relative rotational movements, they cancel each other's swirling effects to produce a high pressure axial output.

22. An axial flow pump/marine propeller according to claim 21, further comprising a set of a second stator member, a second rotor member and second electromagnetic means so that said two driving members are driven to rotate in opposite directions.

23. An axial flow pump/marine propeller according to claim 21 or claim 22, wherein the central one of said two driving members is supported by a pivot bearing arrangement.

24. An axial flow pump/marine propeller according to any of the preceding claims, wherein said means for propelling the fluid includes a conical impeller member fitted to the upstream end of the pump to further promote the axial flow.

25. An electric generator/flow-meter which has a structure as specified in any of the preceding claims.

26. An axial flow pump/marine propeller and/or an electric generator/flow meter constructed substantially as described herein with reference to Figs. 1 and 2, Fig. 3, Fig. 4, Figs. 5A to 7, Figs. 8 to 8D or Fig. 9 of the accompanying drawings.

The
Patent
Office

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Application No: GB 9516653.4
Claims searched: all

Examiner: Ian Philpot
Date of search: 28 June 1996

Patents Act 1977
Search Report under Section 17

Databases searched:

UK Patent Office collections, including GB, EP, WO & US patent specifications, in:

UK Cl (Ed.O): F1C (CFAA); B7V (VBX)

Int Cl (Ed.6): F04D (13/06)

Other: Online WPI (Questel)

Documents considered to be relevant:

Category	Identity of document and relevant passage	Relevant to claims
X	GB 1413835 A (MASCHINENFABRIK AUGSBURG-NURNBERG) see page 2 lines 71-98.	1, 4, 8

X	Document indicating lack of novelty or inventive step	A	Document indicating technological background and/or state of the art.
Y	Document indicating lack of inventive step if combined with one or more other documents of same category.	P	Document published on or after the declared priority date but before the filing date of this invention.
&	Member of the same patent family	E	Patent document published on or after, but with priority date earlier than, the filing date of this application.

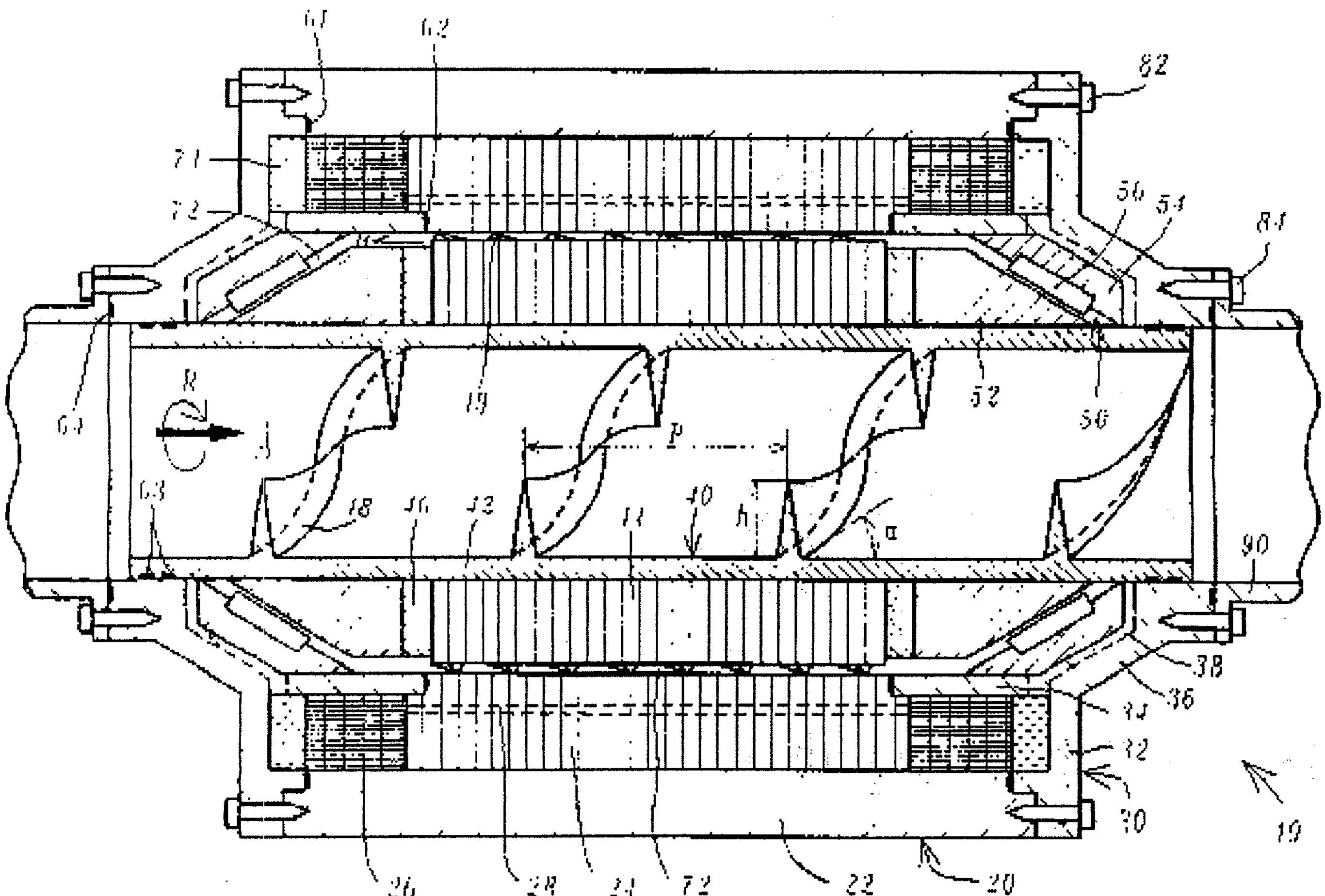
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Fig 1

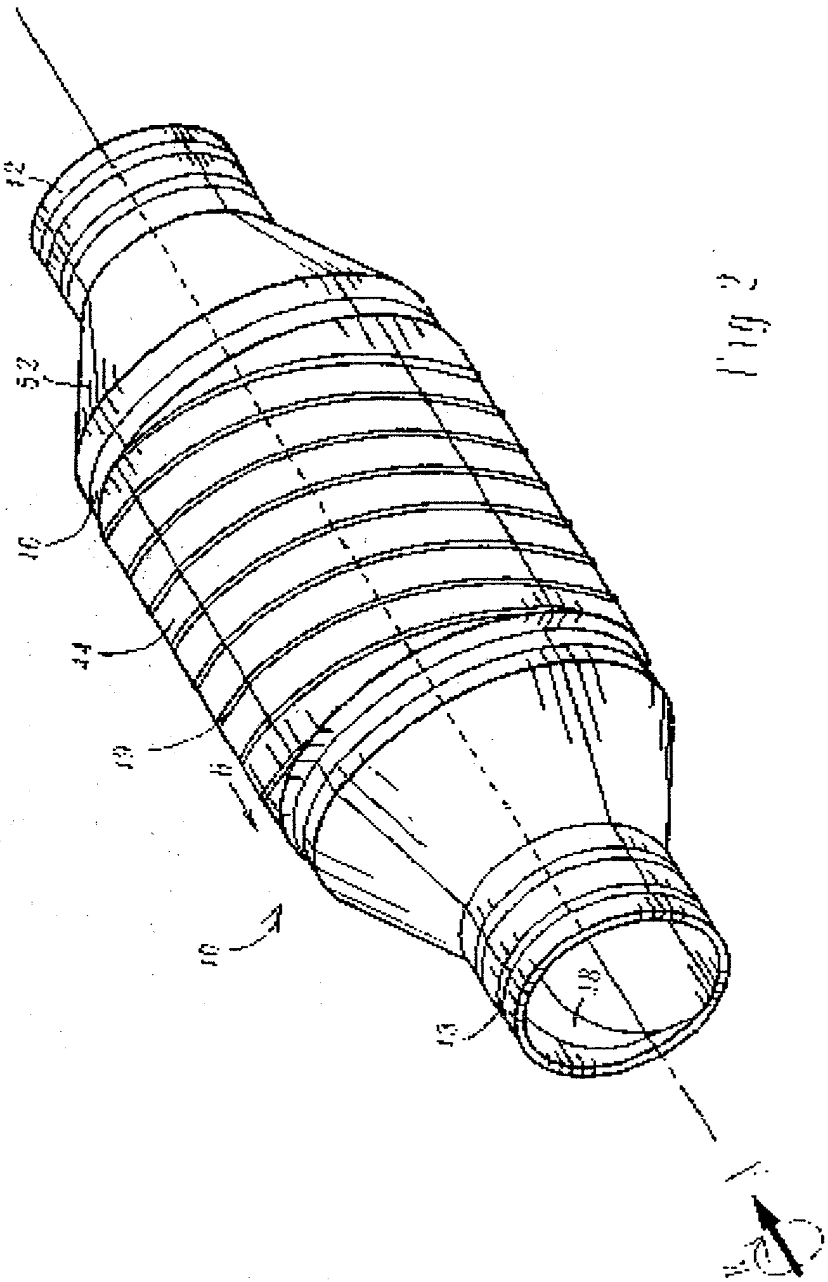
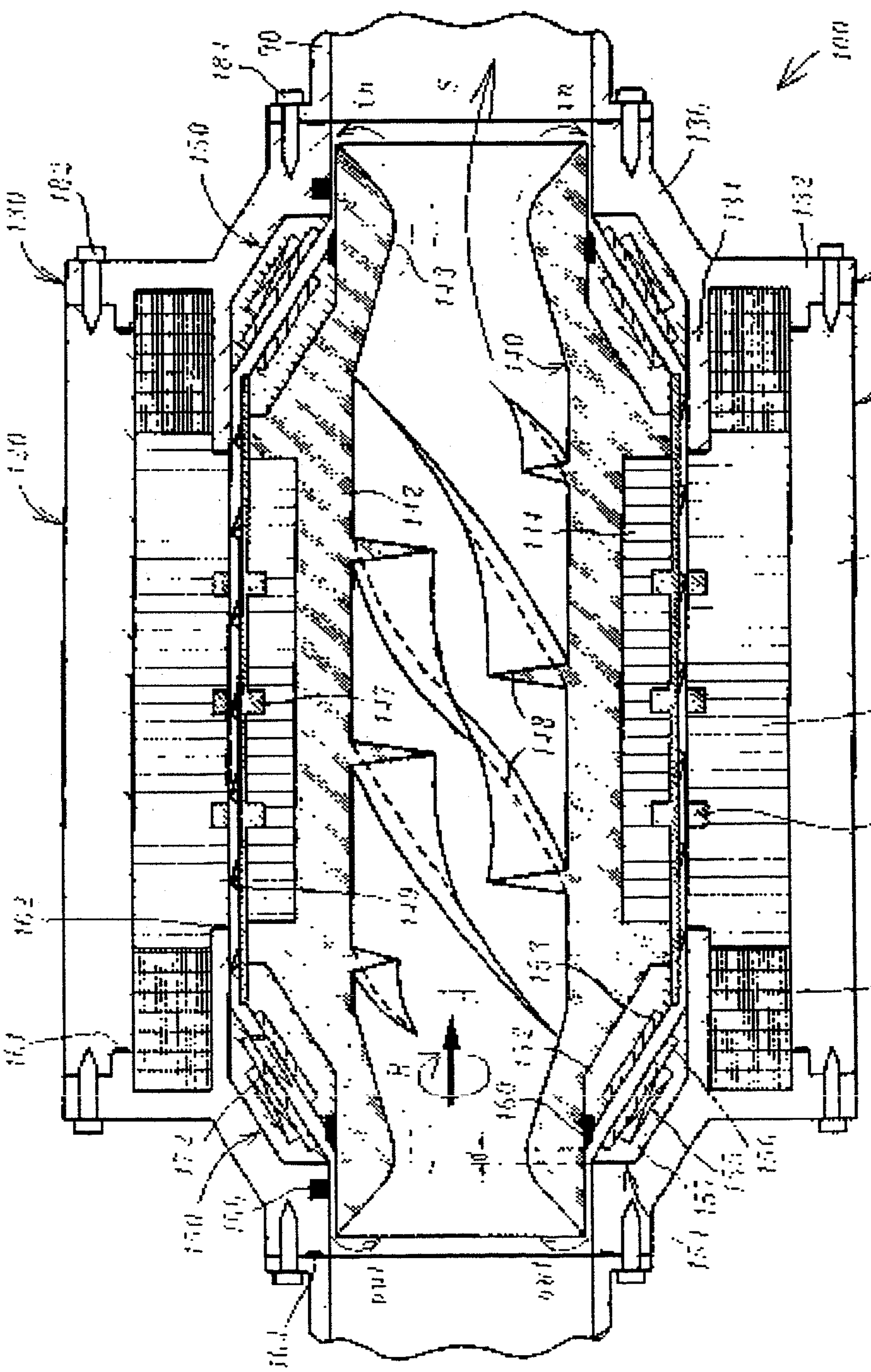


Fig 2



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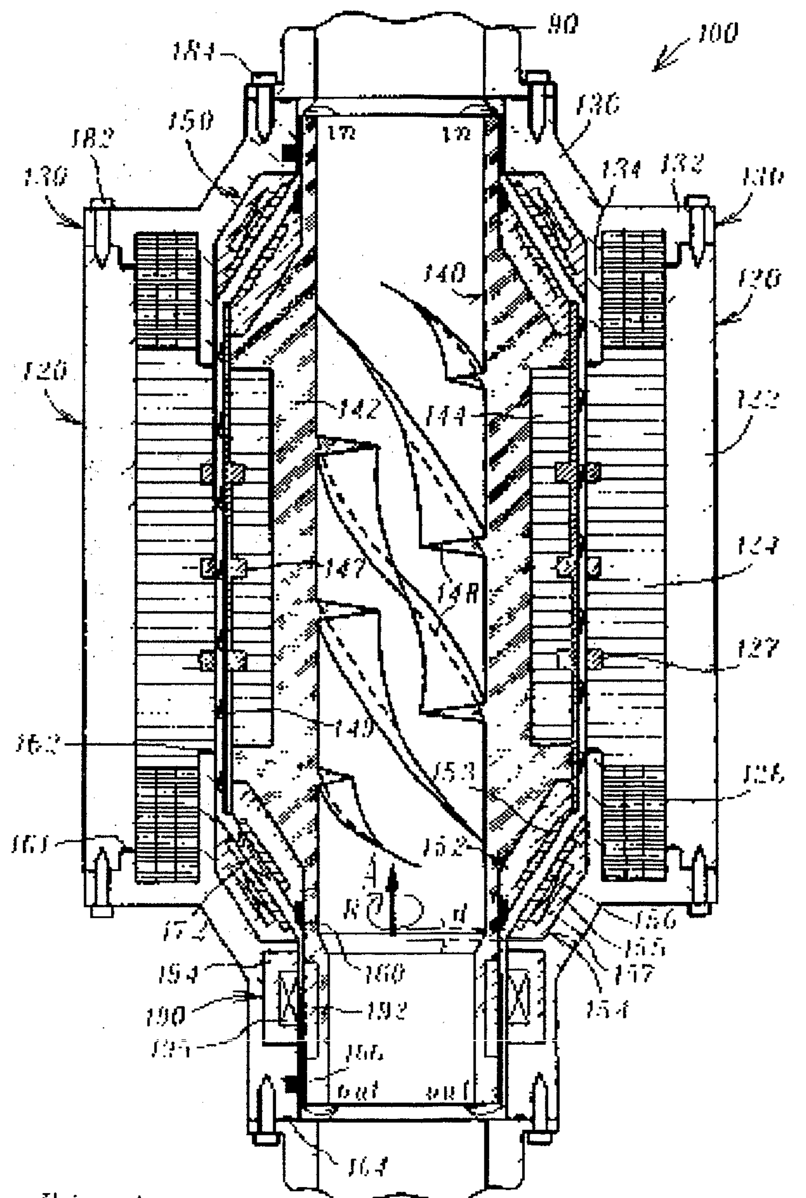
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Fig 4

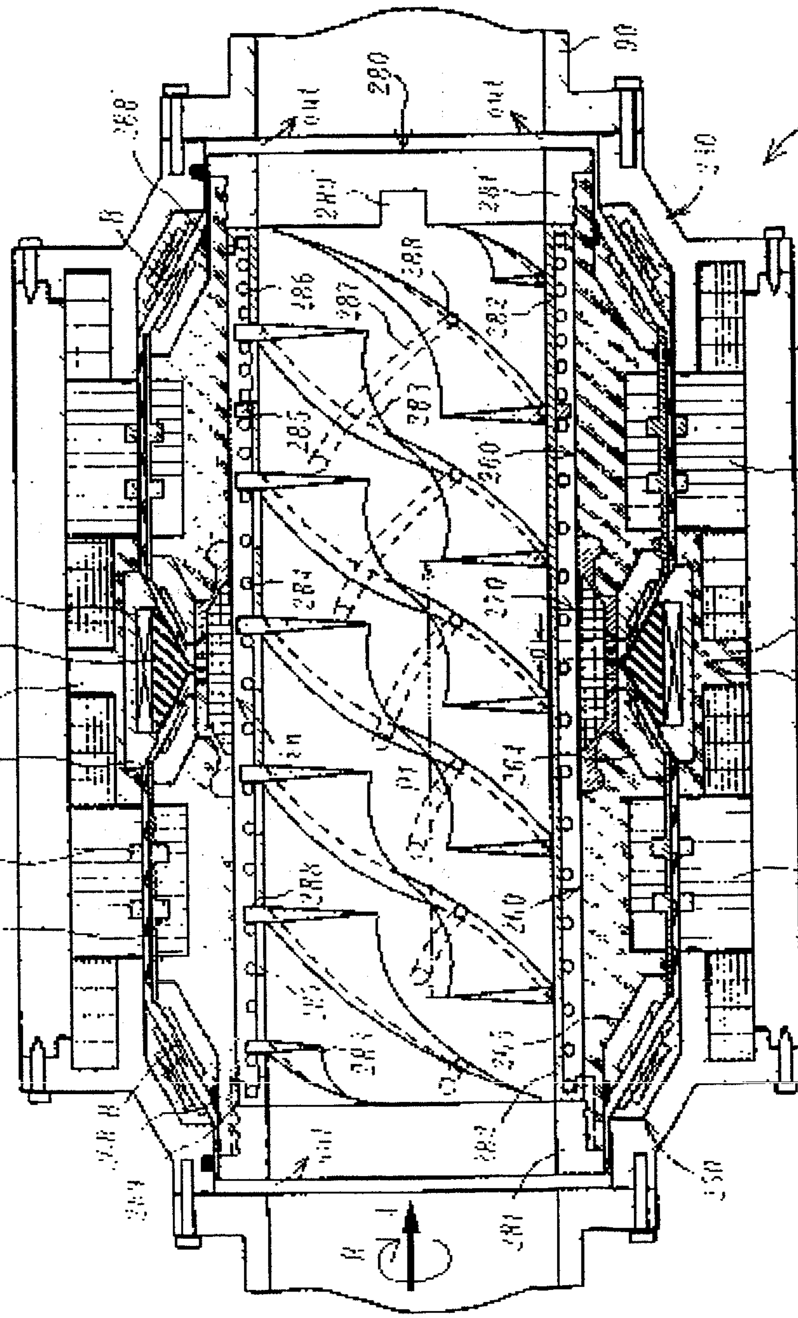

Fig 5a

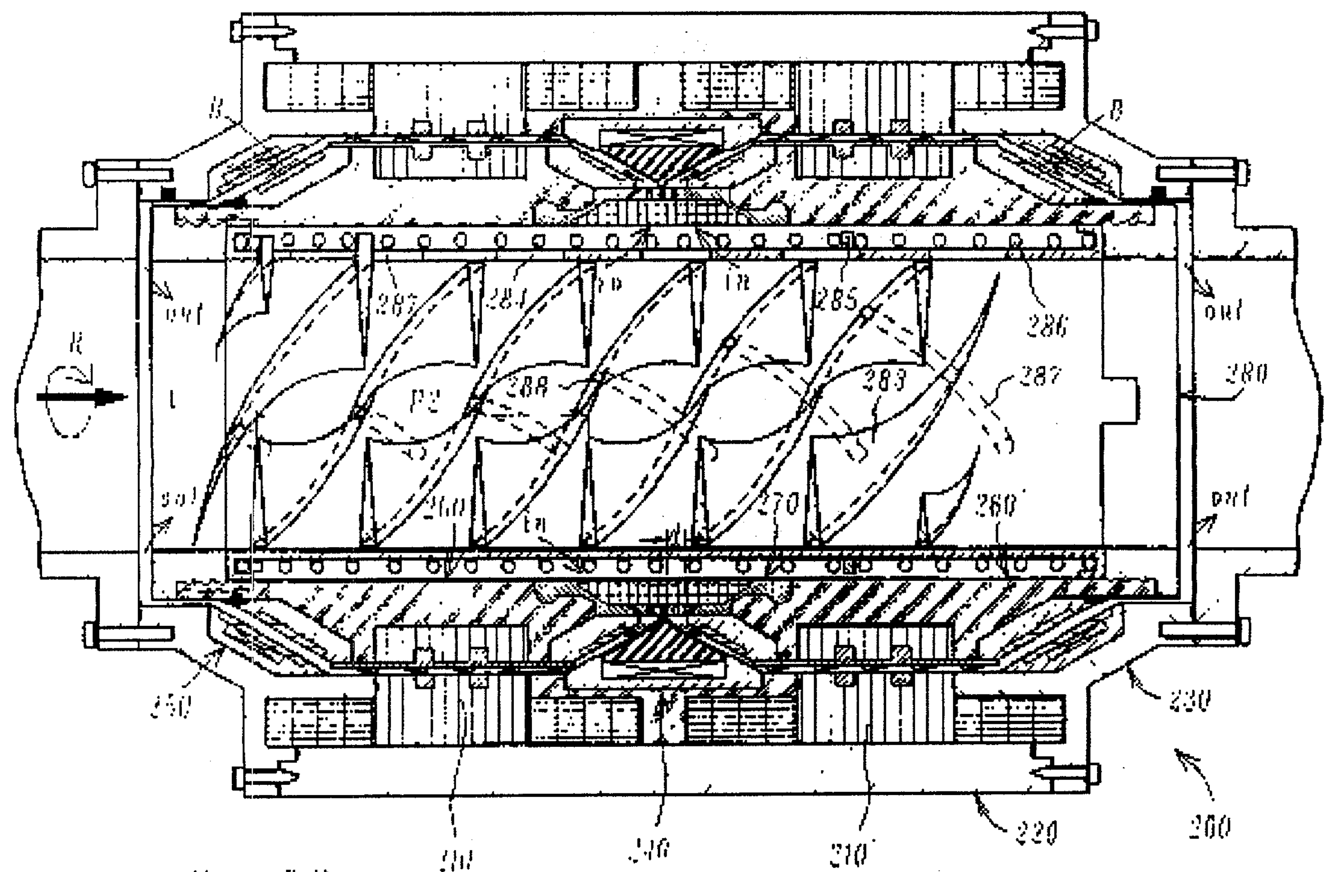
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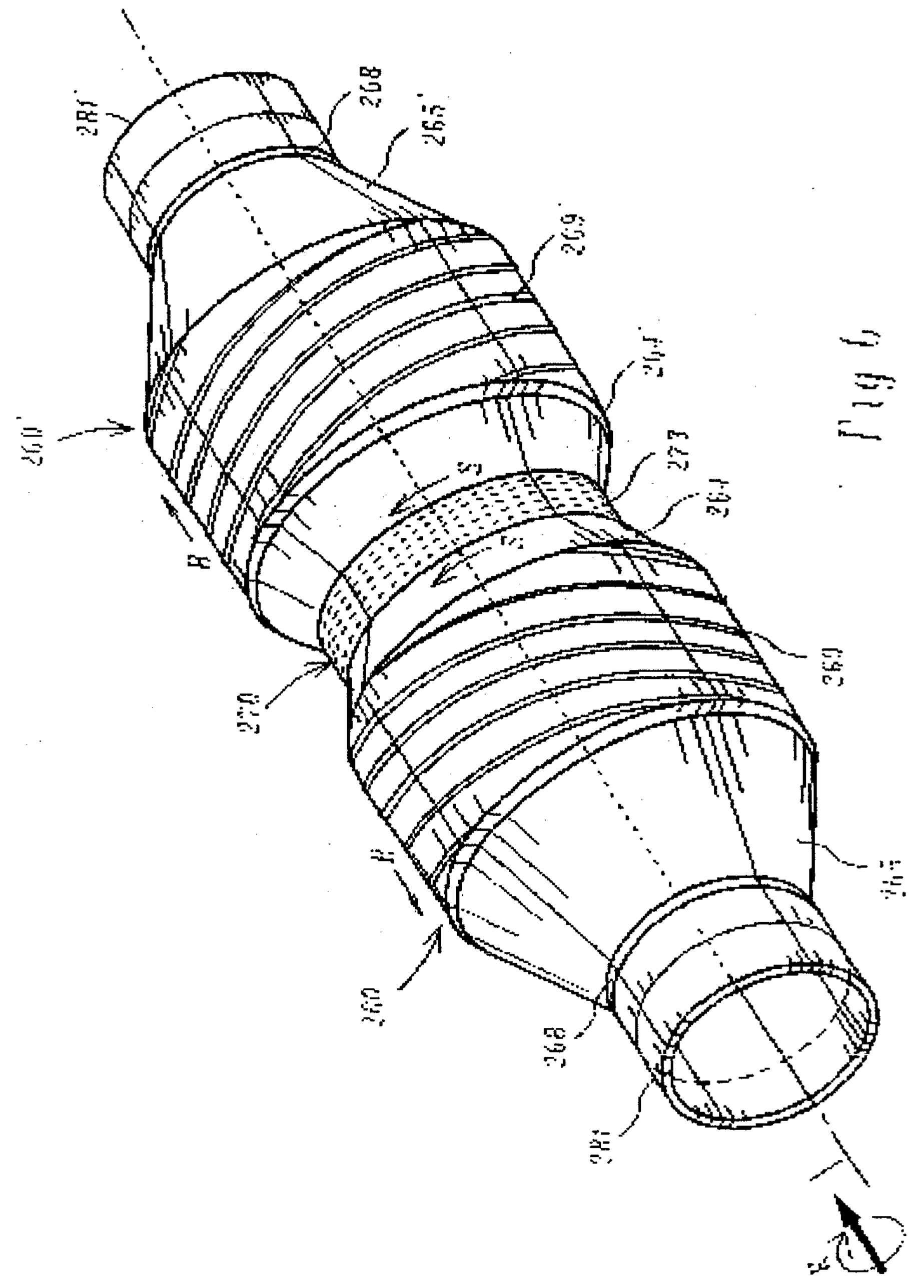
Fig 3B



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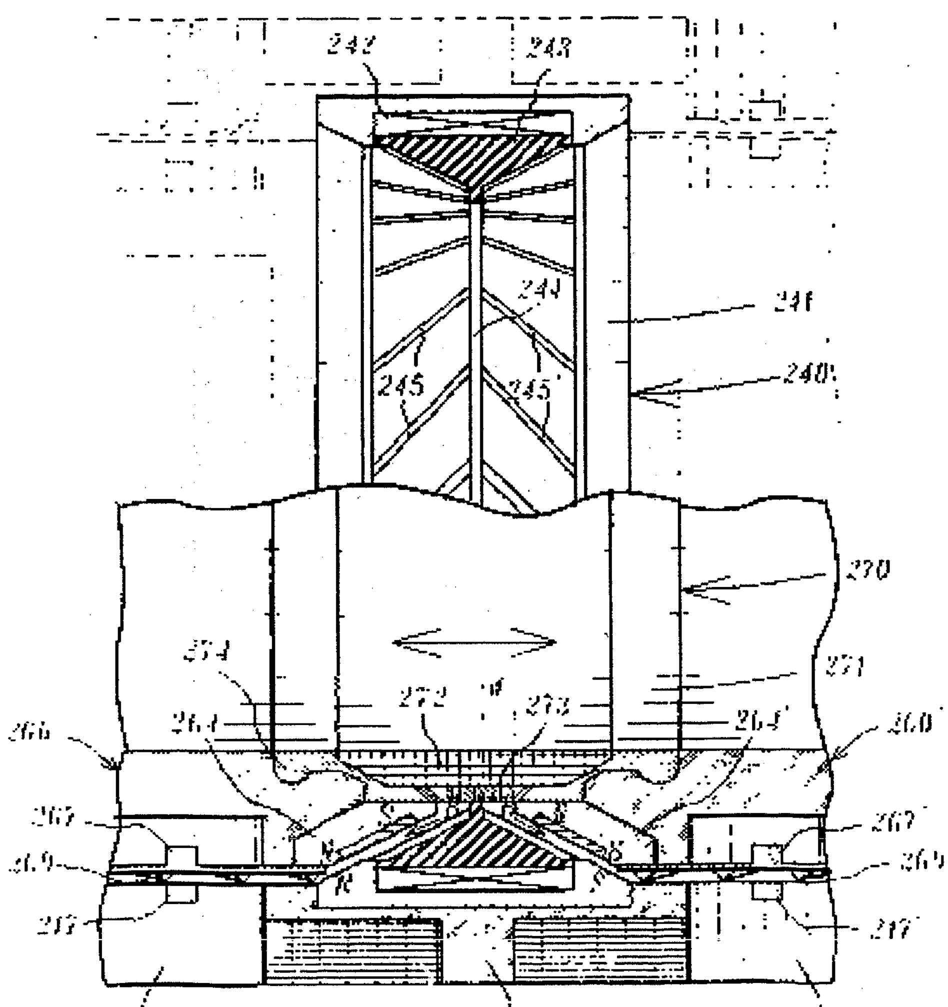
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Fig 7

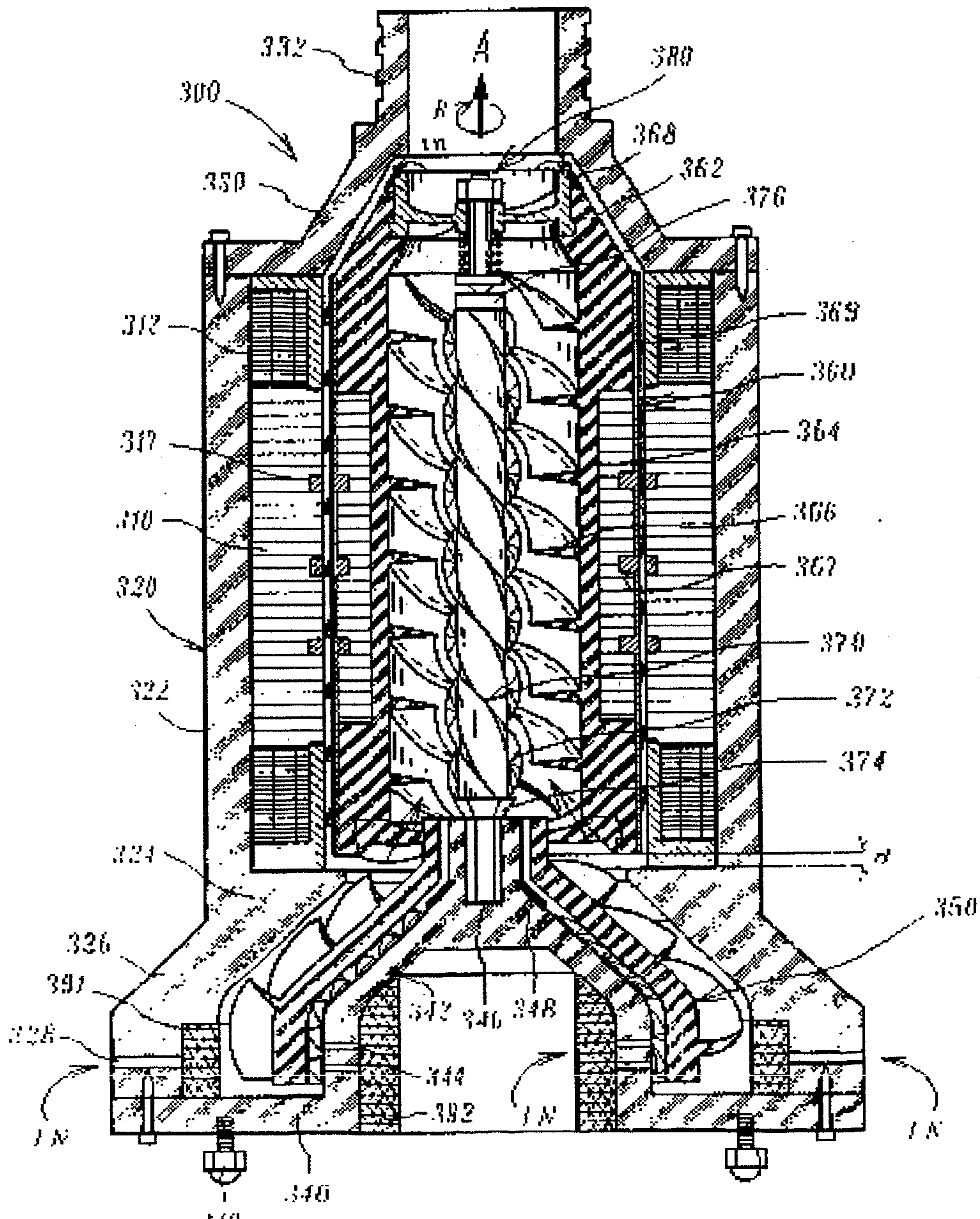

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Fig. 8

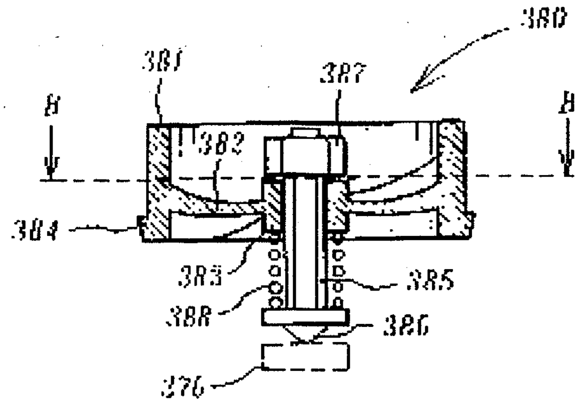

Fig. 8A

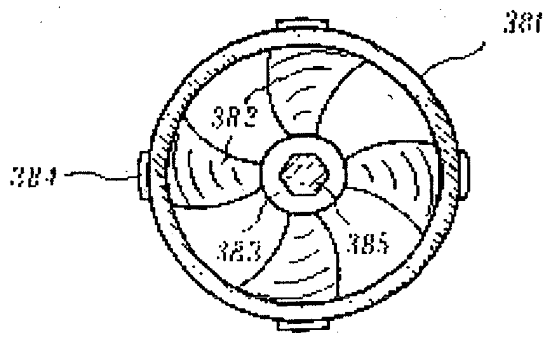

Fig 8B

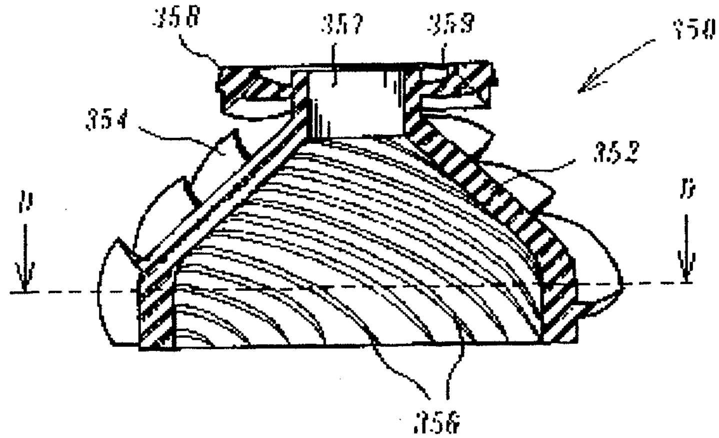

Fig. 8C

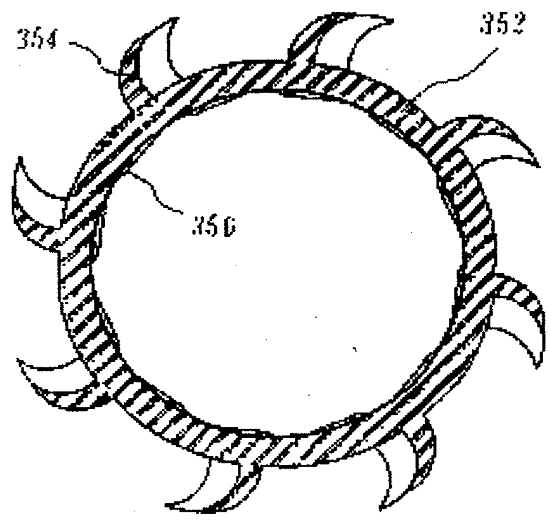

Fig. 8D

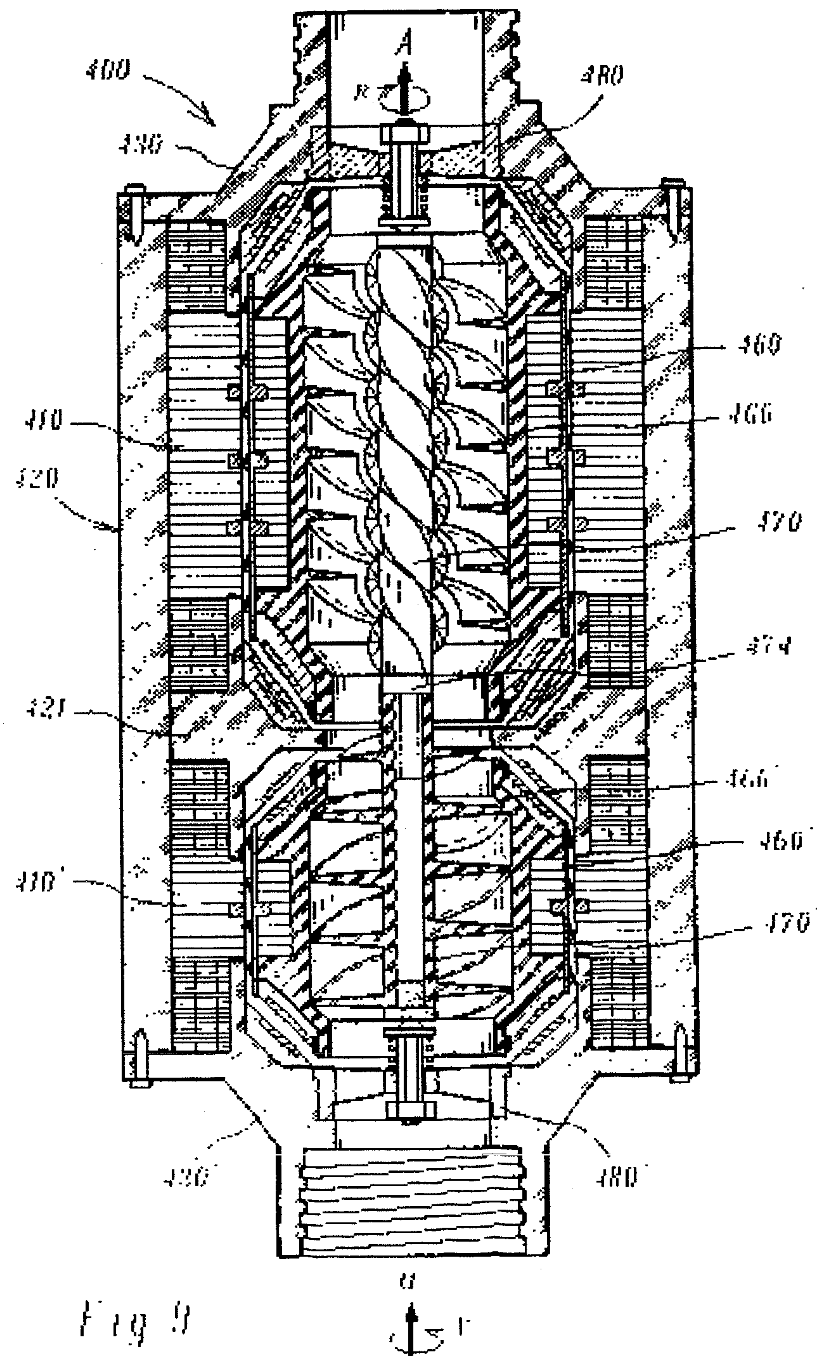
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Fig 9